Marine Engineering

JUNE, 1905.

JUN 8 1905

THE NEW STEAMSHIP BERMUDIAN.

One of the latest steamers to begin service from the port of New York is the twin-screw steel steamship *Bermudian,* belonging to the Quebec Steamship Company, Limited, and operating between New York and Bermuda. This handsome vessel, which we illustrate, was built late in 1904 by Sir James Laing & Sons, of Sunderland, England, and measures 5,530 tons gross. She has a length over all of 435 feet, a length between perpendiculars of 425 feet, a beam of 50 feet, while the depth is 36 feet 6 inches.

THE STEAMSHIP BERMUDIAN.

(Photograph by N. L. Stebbins, Boston.)

One notable feature in the arrangement is the promenade deck, which is 220 feet long and has a clear width without obstructions of 12 feet. This is shown in one of the illustrations. All of the boats are carried on the deck above this, and hence do not interfere in any way. There are 120 state rooms on three decks; those on the shelter deck being nearly all on the outside, and fitted with large windows which are at a sufficient height above the sea to be available for opening in almost any condition of weather. The state rooms on the saloon deck are both inside and outside, the former being lighted and ventilated by means of side ports in the deck-house above them, the deck being cut out in the wake of these lights. The dining saloon, of which we present a photograph, has 154 seats. It extends the entire width of the ship, and is lighted by a handome dome skylight running through the music room, which is also illustrated. The decorative scheme is very handsome, being of hand-carved panels, with ceiling and trimmings of white and gold.

A double bottom extending all the way forward and aft is provided for water ballast, and there is also a deep tank with a water-tight center division, which will contain 400 tons. The two tanks in the double bottom under the engine and boiler spaces have a center-line water-tight division, and are reserved for fresh water, which is used to feed the boilers. The dead-weight carrying capacity of the ship on a mean draft of 20 feet 6 inches is 1,800 tons, which includes 1,200 tons of cargo and 600 tons of coal in the bunkers. There are seven water-tight transverse bulkheads, dividing the ship into eight large water-tight compartments, besides those found in the double bottom. A bilge keel is fitted on each side in order to overcome as much as possible the natural rolling of the ship, and extends for a length of 160 feet. It is

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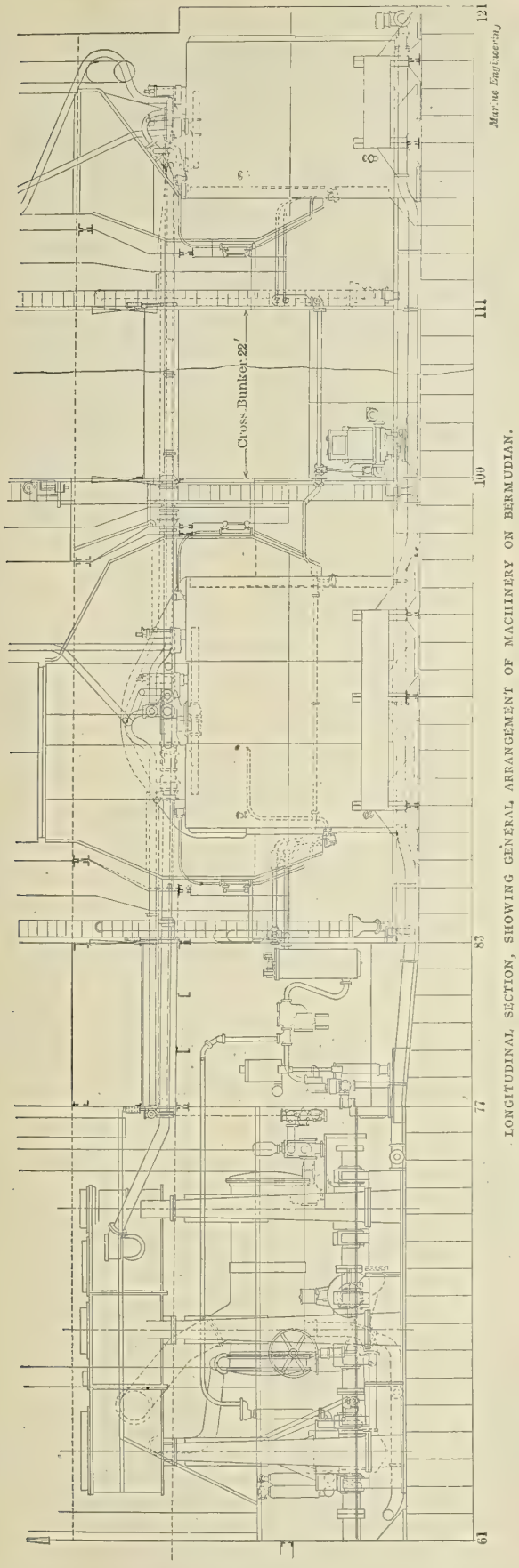
THE MUSIC ROOM ON THE BERMUDIAN.

made of a T-iron 6 by 4 by 1-2 inch with a bulb plate 10 1-2 inches wide and 1-2 inch thick riveted to it.

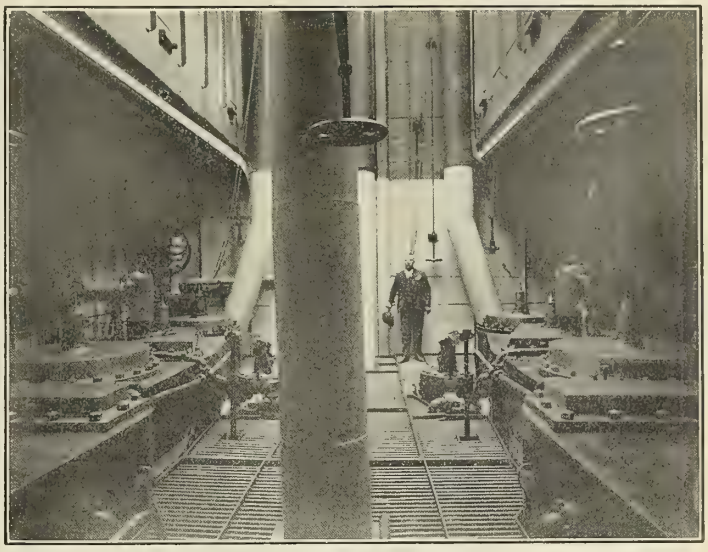
The machinery consists of two sets of triple-expansion engines having cylinders 26, 43, and 71 inches in diameter respectively, with a stroke of 48 inches, the steam being furnished by three double-ended and three single-ended boilers, working under a pressure of 200 pounds per square inch. The single-ended boilers have a length of 11 feet and a mean diameter of 14 feet 9 inches, while the double-ended boilers are 18 feet long with the same diameter as the others. Both forms are fitted with 3 1-4-inch tubes, and have a shell of a thickness of 1 3-8 inches of Siemens-Martin steel. The two funnels rise 96 feet above the grate bars. Each fire-room is fitted with Horace See ash ejectors, which discharge ashes about six inches above the mean water-line.

The crank shaft has a diameter of 14 3-4 inches, with bearings 17 1-2 inches in length. The crank pins measure 15 inches diameter by 17 inches long. The thrust shaft has the same diameter as the crank shaft, and contains eight thrust rings, with a total thrust area for each engine of 1,350 square inches. The propeller shaft has a diameter of 15 inches, and tapers at the rate of 1 inch

THE FIRST-CLASS DINING SALOON.



LONGITUDINAL SECTION, SHOWING GENERAL ARRANGEMENT OF MACHINERY ON BERMUDIAN.



THE UPPER PART OF ENGINE ROOM.

per foot longitudinally where it enters the hub. The propellers measure 17 feet 6 inches in diameter, while the pitch-ratio of 1.23 gives a mean pitch of 21 feet 6 inches. They run at about 90 turns per minute, though on trial they were run up to 103. The requirements for the regular passage between New York and Bermuda call for 16 knots. On the full-power trial trip, which lasted forty-eight hours, the speed obtained with a little under 9,000 horsepower was 18.6 knots.

The entire ship is lighted by electricity and fitted throughout in the most modern manner. The American-De Forest Wireless Telegraph Company have installed their system of telegraphy on the vessel, and an operator dispatches and receives messages for passengers anywhere within 200 miles of Sandy Hook. It is noteworthy that messages sent from this ship when more than 200

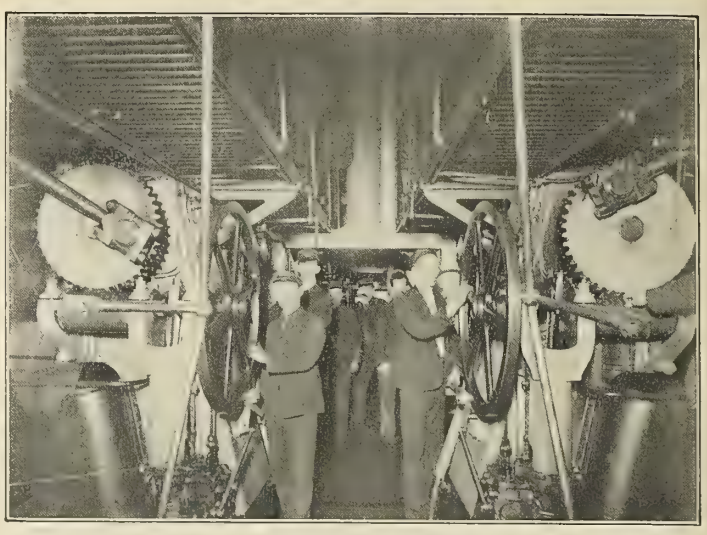
MARINE ENGINEERING DATA CARD. JUNE, 1905.

NAME—Bermudian. **NATIONALITY**—English. **TYPE**—Passenger and freight liner.
LAUNCHED—1904. **BUILDER**—Sir James Laing and Sons. **OWNER**—Quebec Steamship Co., Ltd.
BUILDER OF MACHINERY—Geo. Clark, Ltd., Sunderland.

Length over all	435 ft. 0 ins.
" between perpendiculars	425 " 0 "
Beam, molded	50 " 0 "
Depth, "	36 " 6 "
Load, draft	20 " 6 "
Deadweight cargo capacity	1,200 tons
Gross tonnage	5,530
Coal supply	600 tons
Speed at sea, designed	16 knots
" on trial	18.6 "
Propellers: Two; diameter	17 ft. 6 ins.
Pitch	21 " 6 "
" ratio	1.23
Funnels	2, rising 96 feet above grate

Engines: Type	3-cylinder, triple expansion
Cylinder diameters	26, 43, 71 ins.
Stroke	48 "
Indicated horsepower	8,500
Revolutions per minute	90
Shafting, diameter crank	14 3-4 ins.
" " propeller	15 "
Crank pins	15 ins. dia. × 17 ins. long
Condensing surface	8,650 sq. ft.
Boilers:	3 double-ended; 3 single-ended
Mean diameter, each	14 ft. 9 ins.
Length, d. e., 18 ft. 0 in., s. e., 11 " 0 "	
Working pressure	200 lbs. per sq. in.
Draft	natural

ON THE PROMENADE DECK.



THE STARTING PLATFORM.

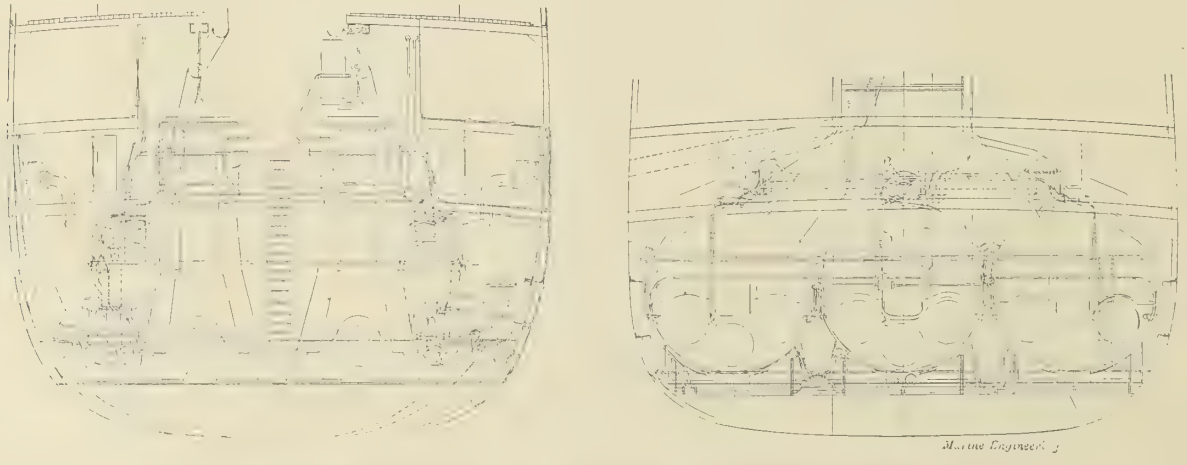
miles out from New York have been clearly read by the De Forest operator at Cleveland, Ohio, and signalled back to the New York office for corroboration.

There are two steam dynamos operated by compound engines and built by the Sunderland Forge and Engineering Company, of England, each of which operates at 300 revolutions per minute and is capable of carrying the entire load of five hundred 16 c. p. lamps. The rated output of the machines is 250 amperes at a pressure of 110 volts. The steering gear is from the works of Alley & MacLellan, of Glasgow. The main feed pumps, which are independent of the engines, are of the Weir type, with cylinders 14 and 10 1-2 inches in diameter and a common stroke of 26 inches. The main condensers, which are fitted well up in the wings, give a vacuum of 27 inches of mercury with a cooling surface each of 8,650 square feet. The refrigerating plant, with a capacity for cooling 15,000 cubic feet of space, has been fitted by J. & E. Hall, of Deptford, and operates on the carbon-dioxide principle. The insulated space actually fitted for service from this machine is only 6,000 cubic feet. The cargo winches fitted on deck are four in number, of friction type, having each two cylinders 8 inches by 10 inches, and were furnished by Williamson Brothers, of Philadelphia. In addition to these there is a large warping winch fitted just forward of the steering-gear house, with cylinders 8 inches by 12 inches, and furnished by Clarke, Chapman & Company, of Newcastle, England.

The ship was built and equipped under the supervision of Mr. P. Baldwin, the company's superintendent. The machinery and boilers were constructed by George Clark, Limited, Sunderland, to specifications prepared by Mr. Henry Black, superintending engineer of the Quebec Steamship Company, under whose inspection they were installed and tested. The vessel is classed 100 A1 in Lloyd's Register of Shipping. The passenger accommodations will take care of 240 first-class, 32 second-class, and 48 third-class passengers. Nearly all of this accommodation is on the upper decks in the vicinity of the music room, social hall, and smoking room.

The *Bermudian* sailed for Bermuda on her maiden trip on the 7th of January. Our photographs were all taken by Mr. N. L. Stebbins, of Boston.

Oil as Fuel on a Battleship.—The four after boilers of the British battleship *Prince George,* now undergoing repairs at Portsmouth, are being fitted to burn either oil or coal, there being two sprays to each furnace for the distribution of the oil. Tanks will be constructed in the double bottom to take 400 tons of oil, auxiliary fuel, to be used in case of emergency, such as the coal having run short or steam being quickly required.



SECTIONS SHOWING INSTALLATION OF ENGINES AND BOILERS ON BERMUDIAN.

LAUNCH OF THE NEBRASKA, OCTOBER 7, 1904.

LAUNCHING OF THE UNITED STATES BATTLESHIP NEBRASKA.

On October 7, 1904, this ship was launched by the Moran Brothers Company, of Seattle, Washington, in whose yard she is now being completed. We are indebted to her builders for the information here given of the particulars of the launch, as well as for the accompanying illustrations. The view of ship alongside dock was taken March 29, 1905.

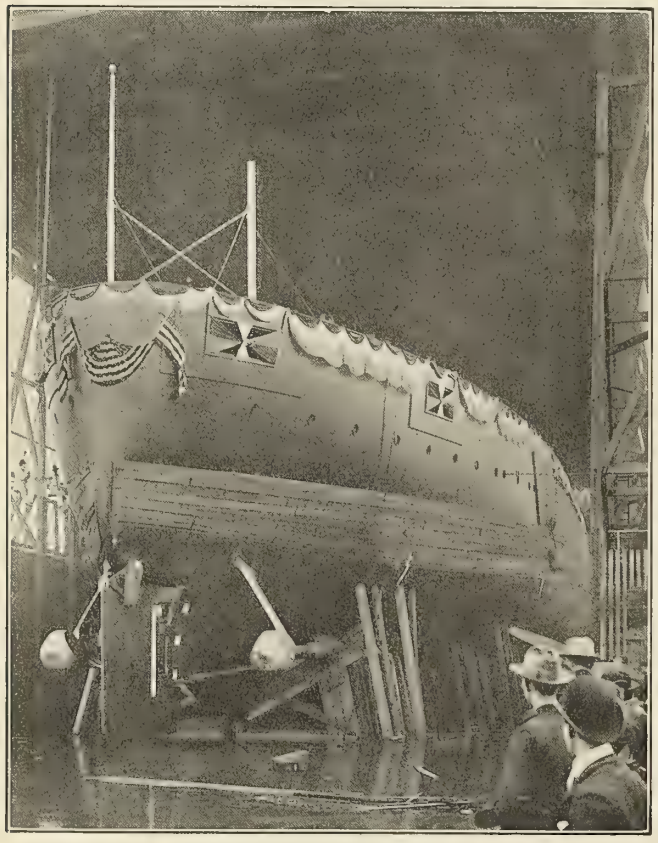
STATE OF COMPLETION.

At the time of launching the percentage of completion of all work pertaining to hull, ordnance, and equipment was 59.4, such percentage being based on the vessel completed under the contract requirements. The machinery on the same date was 79 percent completed. All structural work below the protective deck, except parts affected by recent changes, was completed, and nearly all similar work above the protective deck was in place and riveted. About two-thirds of all stanchions were in place. Openings for all doors were cut, and most of the doors fitted in place. All manholes to inner bottom were fitted and bolted down during the launch, except those in certain 6-inch magazines, where the inner bottom plating was pierced for work in connection with stowage of ammunition. All openings for outboard valves were cut and valves fitted, including main injections and outboard deliveries; openings for the hawse pipes were cut; opening for torpedo tubes not yet laid off. All outside hull work completed, rudder completed and in place; tail-shafts and hubs of propellers in place, zinc and fair-waters on. All deck plating fitted and riveted, lower part of main mast in, bitts and chocks in place, and many of the hatches completed; work on bridges not yet commenced. Main drain was completed and tested, and secondary and auxiliary drains about 90 percent completed. All generating sets, windlass and engine, were installed. Fresh and salt-water system work laid out on ship; fire main work laid out and piping in machinery spaces fitted; air ports in place. Interior casemate armor and the large tubes for the broadside turrets and the signal tube were completed. The broadside barbette armor was in place and the armored ammunition tubes on board. The transverse and longitudinal splinter bulkheads were all in place and almost completed. All wood backing for armor was fitted. A large part of the holes

CONDITION OF THE NEBRASKA, MARCH 29, 1905.

for various pipes in the ship were cut. Gun beds and foundations were about three-fourths completed. All compartments adjacent to the shell plating had been tested, the testing of the water-tight compartments as a whole being over one-half completed. The boilers were installed and up-takes fitted, the ice machine and evaporators, work-shop tools, and most of the auxiliary machinery being in place. In the shop the work was well along on the main engines and various machinery items, superposed turret foundations, masts, forgings for various gears, fittings for various systems, etc. Joiner work also in hand. The weight of the ship at the time of launching was 6,549.95 tons, which was subdivided as follows:

	Tons.
Structural work of hull	4,629.28
Armor	304.50
Splinter bulkheads	83.28
Protective deck, nickel steel	241.86
Wood	143.26
Paints, etc.	65.81
Hull auxiliaries (windlass, etc.)	28.73
Piping, valves, fittings, air ports, etc.	110.19
Equipment weights	52.72
Steam engineering weights	805.32
Miscellaneous dunnage and foreign weights (estimated)	85.00
Total	6,549.95



THE STERN BEFORE THE LAUNCH.

LUBRICANT.

The grease used was put on the ways September 18 and 19, 1904, and consisted of 1,950 pounds of tallow, 1,890 pounds of stearin, and 986 pounds of soap. The day before the launch the after ends of the ground ways were greased with 500 pounds of tallow and 450 pounds of stearin, making a total of 5,776 pounds of grease used in the launching. The first grease applied was pure stearin, then a mixture of stearin and tallow (60 and 40), and final coat of whale-oil soap. It was protected from dirt, etc., by light board strips on each side of ways. To take the weight off the grease the sliding and ground ways were separated by oak distance pieces 5-8 inch thick by 6 inches wide by 46 inches long, spaced about 10 feet apart. When the ship was launched there was no smoking of the ways, the lubricant was left almost intact, and almost 4,000 pounds were removed from ways the day after.

HYDRAULIC JACKS.

To assist in starting the ship one 100-ton hydraulic jack was located on the starboard ground way, and three smaller ones on the port ground way, these three totaling 110 tons. These jacks were butted against fir timbers that were securely bolted to the ground ways, the thrust of the jacks coming squarely against the head of the sliding ways. It was not necessary to use the jacks in starting the vessel. There being plenty of water, no attempt was made to stop the ship in any limited distance.

WEDGES.

During its construction the packing was supported by tapered pieces of fir 2 inches by 4 inches by 46 inches, which separated it from the sliding ways. On the day before the launching these pieces were replaced by fir wedges. These wedges under the poppets were 7 feet long and 9 inches wide, and under the solid packing were 6 inches wide, all being 6 inches thick tapering to 1-8 inch. Under the poppets, both sides, were 92 wedges, and under the solid packing, both sides, were 340 wedges.

The wedges were distributed as follows:

18 wedges 9 inches wide, each side, between frames 11 and 20.
80 wedges 6 inches wide, each side, between frames 20 and 50.
90 wedges 6 inches wide, each side, between frames 50 and 84.
28 wedges 9 inches wide, each side, between frames 84 and 98.
Total number, one side, 216; both sides, 432.

GROUND WAYS AND SLIDING WAYS.

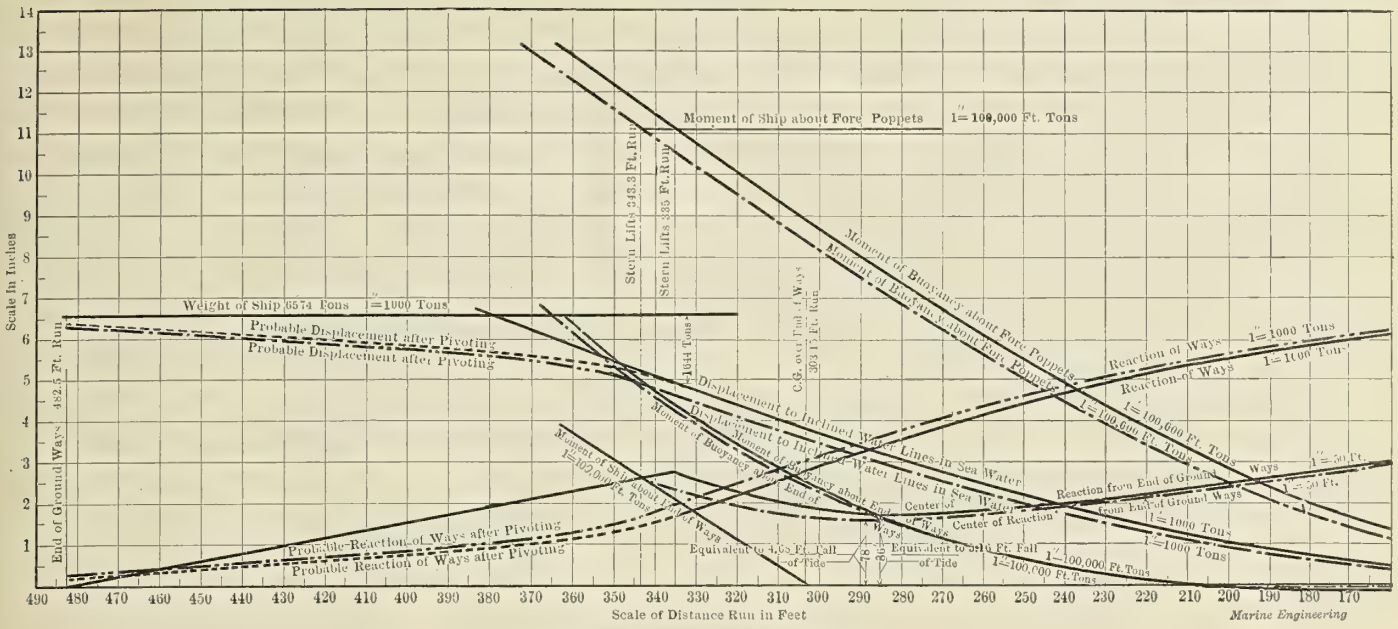
All parts of ground ways were exposed at low water. The total length of ground ways measured from sawing-off stations to ends of ways was 482 feet 6 inches, of which 172 feet 6 inches were under water. The spread of the ways measured between inner faces of the rib-bands was 30 feet 9 inches, the ways being parallel and the surfaces without transverse inclination. The ground ways were inclined 72-100 inch per foot throughout their length. These ways were supported by piles extending out to the ends, the transverse cap pieces being discontinued 16 feet from the end of the ground ways. The depth of water at end of ground ways was 9 feet 8 1-4 inches at time of launching, and 10 feet 4 inches for mean high water. The total length of sliding ways from sawing-off stations was 359 feet 6 inches. There were on each side eight poppets forward, extending from frame 11 to frame 19, and fourteen poppets aft extending from frame 84 to frame 98. The packing under forward poppets was separated from sliding ways by crushing pieces. In fitting the cradle care was taken that no pressure was brought on the laps of the plate ends. The after poppet was 30 feet forward of center of rudder pintles. The after ends of the ground ways for 178 feet were made in two thicknesses the upper part being 4 inches thick. These upper pieces were not fitted until a few days before the launch, thus preventing damage to under-water lubricant.

SOLE PIECES.

The sliding ways and ground ways were tied together at the head of the ways by means of 7-8-inch through-bolts, steel plates 7-16 inch thick top and bottom, acting as washers. Between the sliding ways and the ground ways a key of oak was inserted just forward of the saw cut. Just aft of the saw cut and between the same ways a 3-8-inch thick steel plate, 42 inches wide and 40 feet long, was securely fastened to the sliding ways to prevent freezing. The area of cross section of saw cut for one side was 420 square inches. The ship broke away when a little more than 1-4 of this area was left.

CONDITION OF THE SHORES BEFORE THE LAUNCH.

On October 6, 1904, the day before the launch, the ship was supported as follows: between the keel and ground ways and ex-


Moment of Ship about Fore Poppets 1"=100,000 Ft. Tons
Moment of Buoyancy about Fore Poppets
Weight of Ship 6574 Tons 1"=1000 Tons
Probable Displacement after Pivoting
Displacement to Inclined Water Lines in Sea Water
Reaction of Ways
Probable Reaction of Ways after Pivoting
Scale in Inches
Scale of Distance Run in Feet
Marine Engineering

LAUNCHING CURVES FROM NEBRASKA.

tending from frame 12 to frame 89 there was a line of shores spaced four feet apart; fifteen feet outboard of ground ways there was a line of shores extending from frames 25 to 90 and spaced 4 feet apart; twenty-three feet outboard of ground ways there was a line of shores extending from frame 21 to frame 89 and spaced four feet apart. Total, one side, 213 shores; both sides, 426 shores. Besides the above-mentioned shoring, the ship was supported by cribbing under docking keels at frames 37, 43, 48, 56, and 70.

SHORING IN SHIP.

Over the line of the launching ways, extending from frame 11 to frame 18, the ship was shored as follows: on frames 11 and 12, port and starboard, and extending from second longitudinal to center of ship, there were two 8-inch by 8-inch diagonal shores; between frames 13 and 18, port and starboard, and extending from second longitudinal to lower platform, and between lower and upper platforms, there were lines of 8-inch by 8-inch shores. In each fire-room there were two 8-inch by 8-inch shores placed on blocking extending between second and third longitudinals, the shores being located between the boilers, all of which were in place.

MEASUREMENTS TAKEN BEFORE AND AFTER LAUNCHING TO DETERMINE HOG OR SAG.

For observing the hogging and sagging strains a transit instrument was located at frame 8, starboard side, and 10 feet 9 inches from center line, on a platform 4 feet high. At 105 3-4 starboard, a batten was erected on which was placed a target 13 feet above deck and 10 feet 9 inches from center line. Along either side of this line of sight three battens were located on which were read directly the deflections. Batten No. 1 was located on forward end of superstructure 10 feet from center line; batten No. 2 at frame 53, 10 feet 3 inches from center line; and batten No. 3 on after end of engine hatch coaming 11 feet 3 inches from center line. Just before any wedging-up had been done the scales on the three battens were set with zero in the line of sight. No appreciable deflection occurred until all the blocks and shores had been knocked out and the ship had settled on the ways, when the following readings were taken:

Batten No. 1 3-32 inch,
Batten No. 2 1-4 inch,
Batten No. 3 5-32 inch,
} Sagging.

After the ship was launched the readings were as follows:

Batten No. 1 1-8 inch,
Batten No. 2 1-4 inch,
Batten No. 3 3-16 inch,
} Hogging.

DIMENSIONS.

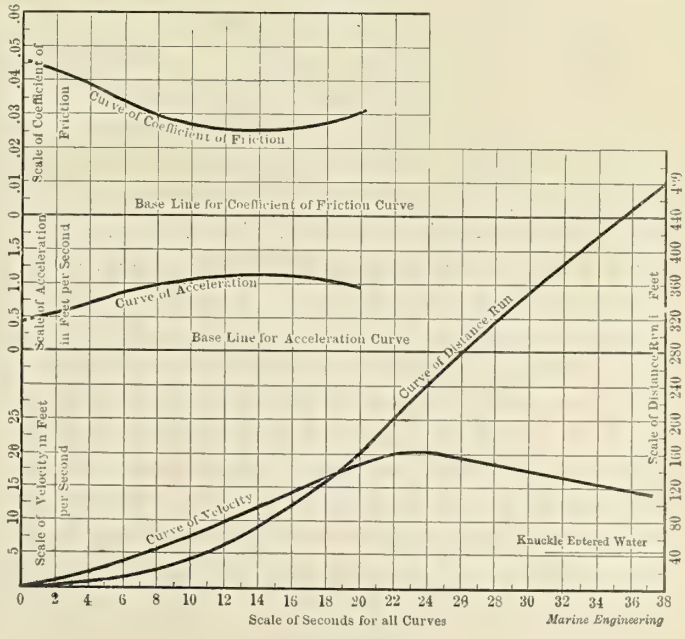
Length over all	441 ft. 3 ins.
Breadth (extreme)	76 ft. 2 1-2 ins.
Depth (keel to main deck at side), molded	41 ft. 4 1-2 ins.
Percentage of completion at launching	59.4

MEN EMPLOYED ON SHIP.

Caulkers	4
Fitters	4
Riveters	2
Machinists	5
Plumbers	6
Ship riggers	20
Government employees	5
Total	46

MEN EMPLOYED OFF SHIP.

Ship carpenters	91
Yard laborers	14
Boat-yard laborers	22
Ship-shed laborers	59
Electricians	2
Repair Department	4
Total	192


Curve of Coefficient of Friction
Base Line for Coefficient of Friction Curve
Curve of Acceleration
Base Line for Acceleration Curve
Curve of Distance Run
Curve of Velocity
Knuckle Entered Water
Scale of Seconds for all Curves
Marine Engineering

LAUNCHING CURVES.

LAUNCHING DATA.

Inclination of building slip per foot, 3-4 inch.
Inclination of ship per foot, 5-8 inch.
End of sliding ways below mean water level (measured along ways), 47 feet.
Length of sliding ways, 359 feet 6 inches.
Width of sliding ways, 3 feet 6 inches.
Thickness of sliding ways, 10 inches.
Area of sliding ways, square feet, both sides, 2,516.5.
Elevation of after knuckle above mean high water, 1.95 feet.
Launching weight of ship (weight record), 6,464.95 tons.
Launching weight, dunnage and shoring (estimated), 85 tons.
Weight of ship as launched (sum of above), 6,549.95 tons.
Launching cradle, including sliding ways (calculated), 169.25 tons.
Launching weight, total, 6,719.2 tons.

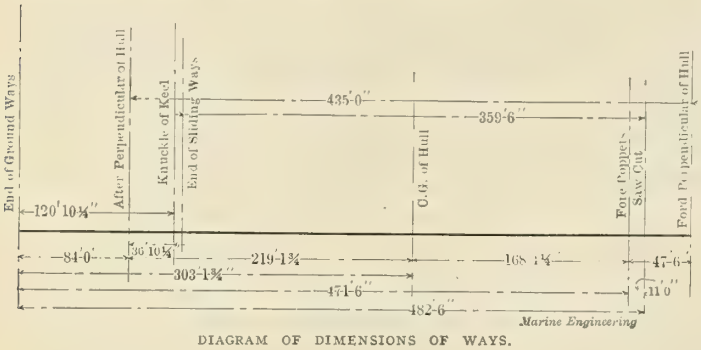


DIAGRAM OF DIMENSIONS OF WAYS.

Pressure on ways, tons per square foot, 2.67.
Draft forward, 11 feet 3 inches.
Draft aft, 12 feet 6 inches.
Mean launching draft, 11 feet 10 1-2 inches.
Corresponding displacement, water density 1.013, 6,515 tons.
Pressure on fore poppets at pivoting (calculated), 1,643.8 tons.
Time of starting to maximum velocity, 23 1-5 seconds.
Maximum velocity, feet per second (measured along ways), 20.25.
Maximum velocity, knots per hour (measured along ways), 11.99.
Distance traveled to maximum velocity (measured along ways), 224 feet.
Coefficient of friction, initial, .046.
Distance traveled to pivoting point (calculated), 335 feet.

THE TUG BOATS WM. E. CLEARY AND J. H. WILLIAMS.

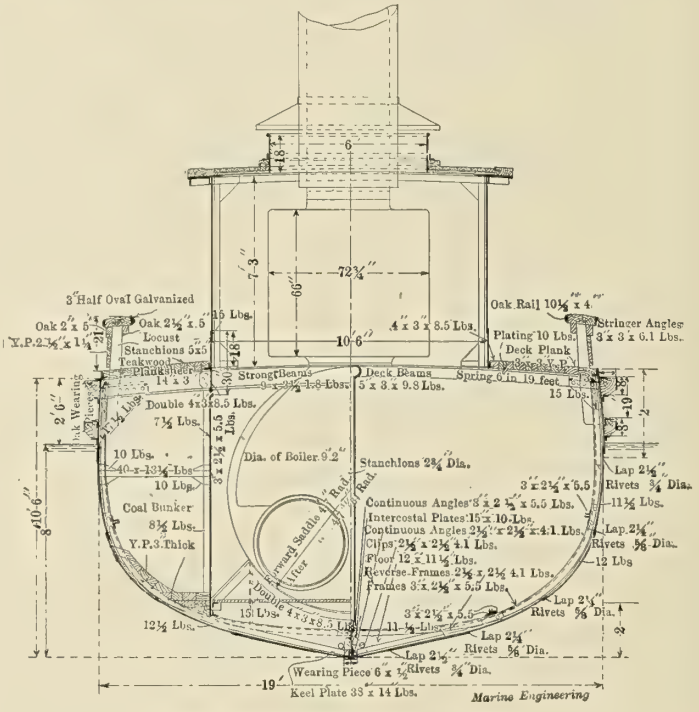
The tug boats *William E. Cleary* and *J. H. Williams*, owned and operated by the Cornell Steamboat Company, were designed by Mr. J. A. Hargan of this city, and built by T. S. Marvel of Newburg, N. Y. These tugs are built of steel, having a length between perpendiculars of 65 feet 6 inches, and a length over all of 75 feet. The molded beam is 19 feet and the depth at side 10 feet 6 inches. This gives a depth of hold of 8 feet 7 inches. The draft under ordinary conditions is 8 feet, but when carrying the full load, 15 tons of coal and 6 tons of water, the maximum draft aft becomes 9 feet 7 inches. The freeboard amidships is 2 feet 6 inches, the normal displacement 143 tons, while the net and gross tonnages are 61 and 87 tons respectively.

The hull is divided into five water-tight compartments separated by bulkheads located on frames 4, 8, 11 and 33. The frames consist of steel angles measuring 3 by 2 1-2 inches by 5.5 pounds, and are spaced 22 inches apart. The deck beams, which occur at every frame, are steel angles measuring 5 by 3 inches by 9.8 pounds. A flat plate keel is fitted 38 inches wide and 14 pounds in thickness. The stern post is of wrought iron 5 1-2 by 2 1-2 inches, and the stem, of the same material, is 5 1-2 by 1 1-2 inches. The shell plating, which is of steel, has a thickness of 11 1-2 pounds, except that the bilge strake is 12 pounds thick and the shear strake 13 1-2 pounds. The deck planking consists of yellow pine measuring 3 by 3 inches.

There is one condensing engine with a single cylinder having a diameter of 16 1-4 inches and a stroke of 20 inches. This engine develops 245 indicated horsepower when towing, and a maximum of 315 when running light. A piston valve is fitted and admission is cut off at 15 inches when in full gear. The pumps are of the Blake pattern and are all independent.

There is one boiler of the Scotch, single-end type, which has a length of 11 feet 6 inches and a mean diameter of 9 feet 2 inches. The average steam pressure to be carried is 125 pounds per square inch, and the expected gradient between boiler and engine is only one pound. The total heating surface in the boiler is 1,248 square feet, and the grate area is 48 square feet, giving a ratio of heating surface to grate area of 26 to 1. There are two furnaces of the Morison suspension type, having an inside diameter of 36 inches each. The maximum consumption of fuel is about 720 pounds per hour.

There is one cast iron propeller with a diameter of 7 feet and a pitch of 10 feet 6 inches, this giving a pitch-ratio of 145. The propeller is fitted with four blades of a maximum width of 37 inches, and is from the shops of T. S. Marvel.


3" Half Oval Galvanized
Oak Rail 10½" x 4"
Stringer Angles 3" x 3" x 6.1 Lbs.
Plating 10 Lbs.
Deck Plank
Deck Beams 5 x 3 x 9.8 Lbs.
Spring 6 in 19 feet
Double 4 x 3 8.5 Lbs.
Dia. of Boiler 9'2"
Stanchions 2¾" Dia.
Coal Bunker
Y.P. 3 Thick
Continuous Angles 3" x 2½" x 5.5 Lbs.
Intercostal Plates 15" x 10" Lbs.
Continuous Angles 2½" x 2½" x 4.1 Lbs.
Clips 2½" x 2½" 4.1 Lbs.
Floor 12" x 11½" Lbs.
Reverse Frames 2½" x 2½" 4.1 Lbs.
Frames 3" x 2½" x 5.5 Lbs.
Rivets ⅝" Dia.
Lap 2½"
Wearing Piece 6" x ½"
Keel Plate 33" x 14" Lbs.
Marine Engineering

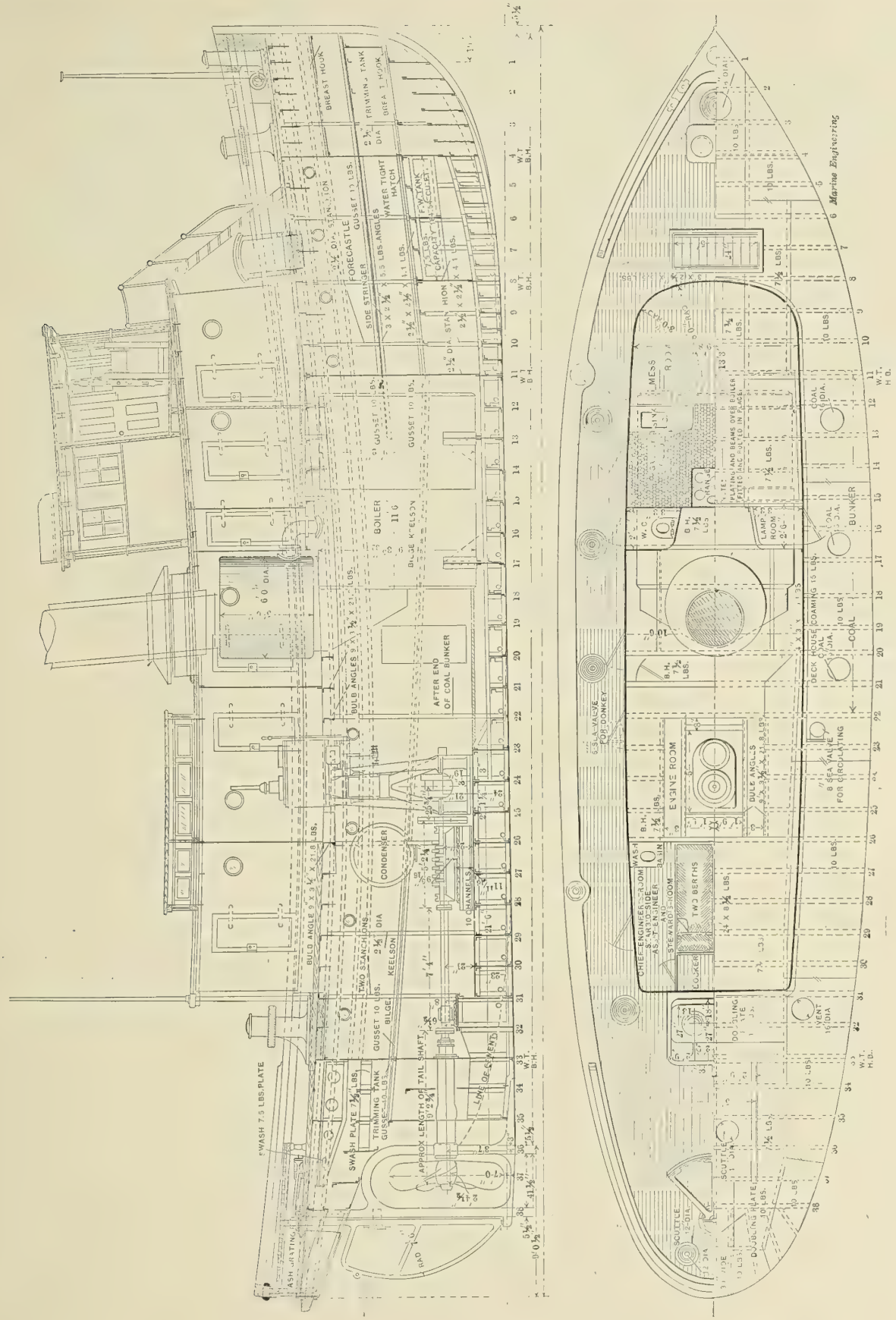
MIDSHIP SECTION OF THE WM. E. CLEARY.

The *Cleary* underwent a test on October 25, 1904, with the following results:

	Mean Cut-off	Steam Pressure	Vacuum.	R.P.M.	M.E.P.	I.H.P.
Running light	15"	125	26"	136	115	315
Heavy towing	15	125	26	106	115	245
" "	12	125	26.5	100	107	216
" "	10	125	27	96	97.5	189

When towing, the revolutions of the engine and propeller are 106; when running light 136. The speed in the latter case is 13 miles per hour. The boats cost $30,000 each, and the fuel capacity is 15 tons. The machinery was built by the Cornell Steamboat Company at Rondout, N. Y.

The Bureau of Navigation report that, during the quarter ending March 31, there were built in the United States 159 vessels of a total of 62,744 gross tons, of which 143 of 10,942 tons were of wood and 16 of 51,802 tons of steel. The steel ships were all



propelled by steam, while of the wooden ships 40 of 6,743 tons were propelled by sail and the balance by steam. The total compares with 177 vessels of 55,056 tons for the corresponding quarter of 1904. For nine months, ending March 31, 1904, there were built 748 vessels of 232,133 gross tons, or an average of 312 tons each; while during the corresponding nine months this year the total number built was 701, of an aggregate of 155,342 tons, or an average of 221 tons each. It is thus seen that not only is the total amount built during the nine months a smaller quantity than it was last year, but the vessels are of a considerably smaller size, and, therefore, the tonnage represents a much smaller effective addition to our shipping.

AMSLER'S MECHANICAL INTEGRATORS.

BY CECIL H. PEABODY.

Several years ago a series of articles was printed in Marine Engineering,* giving a simple geometrical explanation of Amsler's polar planimeter and some other forms of planimeters; the writer now desires to extend the method to the explanation of Amsler's integrator for determining moments, and moments of inertia of plane figures, with the hope that it will be possible to give a good conception of this instrument, which is now so widely used, to any who may find some confusion of thought in explanations that combine the statement of the mechanical construction of the instrument with an integration in the conventional form of the infinitesimal calculus. It is true that the idea of integration must ultimately be introduced, but it may be reserved until the construction and manner of use of the instrument is mastered, and that idea may be so presented that it is comparatively easy of comprehension. It is true, also, that a proper conception of the moment and moment of inertia of a plane figure requires a conception of integration, but that conception does not for our present purpose demand a knowledge of the notation and contrivances of the infinitesimal calculus. The mathematical forms in this article will not require more than the equations of plane trigonometry. The primary steps of the previous discussion of the planimeter will be here repeated, and applied to the instrument under consideration, which determines areas as well as the properties specified.

Fig. 1 gives a photograph of the simplest form of the integrator which can be used to determine areas and moments only; it is also the most satisfactory form because the determination of moments of inertia, for reasons to be pointed out later, is likely to be subject to comparatively large errors. Let PQR (Fig. 2) be an irregular figure on a piece of drawing paper for which we wish to determine the area and the moment with regard to the axis xx'. First we place a straight grooved steel track TT' at the proper distance from the axis xx' by aid of the two trams Tt and $T't'$. The trams are then laid aside so as not to interfere with the use of the instrument, which consists of two rigid arms (with appurtenances) hinged at H. This is placed on the paper with the wheels of the guiding carriage C in the groove in the track. The tracing arm HP is used for following the outline of the figure PQR, as with the ordinary polar planimeter. In fact, the instrusent is a form of rolling planimeter, and it will simplify our discussion to ignore the other appurtenances for the moment, and to discuss it as a planimeter only. We have then the guiding arm CH held at right angles to the groove in the track by its carriage C, so that the hinge H moves on the line xx'; and the tracing arm HP, which carries the area wheel W_a. It must be pointed out in passing that the trams Tt and $T't'$ are of exactly the same length as the arm CH, measured from the groove to the hinge; this setting of the instrument is an essential feature for determining moments, but not for areas. The area wheel is carried on an axis parallel to the tracing arm, at any convenient place on that arm; the only essential features are this parallelism, the diameter or circumference of the wheel, and the length of the arm from hinge to tracing point. Near the tracing point is a foot to carry the weight of the instrument at that place; while the frame that carries the area wheel has an elastic connection which allows the wheel to follow the paper even if its surface is slightly irregular. A tracing point shown at the mid-length of the arm in Fig. 1 can be used for tracing small figures.

Suppose now that the tracing arm were removed from the instrument and held in the hand at one side HP of a rectangle having the same length; if the arm were moved parallel to HP over the rectangle to hp the wheel would roll the path $W_a w'$ equal to Hh. The area of the figure is clearly:

$$HP \times Hh = HP \times W_a w'.$$

The instrument represented by Fig. 1 has the tracing arm eight inches long from hinge to tracing point P and the circumference of the wheel is two and a half inches; the diameter is consequently 0.7957 inch. If the width of the figure is two and a half inches the wheel will make one turn during the operation indicated, and consequently one complete turn of the area wheel measures:

$$2.5 \times 8 = 20$$

square inches. The scale on the wheel is divided into one hundred parts and can be read to one-tenth of a part by aid of the vernier; the number of turns is recorded on a dial near the wheel. In using the instrument, it is customary to read the dial, wheel, and vernier before tracing the figure PQR, and again after the tracing, and then to multiply the difference of readings by the factor for area; in this case by 20. For example:

Final reading..... 3.754
Initial reading..... 2.168

Difference..... $1.586 \times 20 = 31.72$ square inches.

But the same method of measurement may be applied to a parallelogram like Fig. 4, for while the arm moves from HP to hp the wheel will roll and slide over the path $W_a w'$; the sliding does not enter into the determination of the area, and may be ignored; the wheel rolls an amount equal to $W_a w'$, which is equal to the altitude of the parallelogram; consequently the area can be found as before by multiplying the difference of readings by the instrumental factors.

In proceeding with the discussion it is convenient to consider the measurement of the area of a special figure like $pqrs$, Fig. 5, which is bounded by two straight lines pq and rs (of which the latter coincides with the axis xx') and by the two circular arcs ps and qr drawn from centers h and h' on the axis xx', with the arm hp for a radius. The reason for selecting this particular figure will be apparent when we come to the discussion of moments and moments of inertia; if we were considering areas only, a somewhat greater freedom of selection would be possible. It may be noted further that the area of the figure $pqrs$ is equal to that of the rectangle $pqmn$, made by dropping perpendiculars from p and q to the axis xx', for the area of psn is equal to the area of qrm; the moments and moments of inertia about the base xx' are also equal, a fact of importance in the discussion of those properties.

Suppose that the integrator is properly mounted on its track TT', with the hinge on the axis xx', and that it is used to trace the figure $pqrs$. There will be four operations: (1) the tracing arm will move parallel to itself from hp to $h'q$; (2) the arm will swing through the angle $qh'r = \alpha$; (3) the arm, which now coincides with the axis xx', will move along that axis to hs; (4) the arm will swing through the angle $shp = \alpha$ to its initial position. Of these four operations, only the first will produce a permanent effect on the measuring wheel W_a. During that operation the wheel will roll and slide over the path wu, the rolling being equal to

$$wu \sin wuv = d \sin \alpha,$$

where d is the length pq. The area recorded by the wheel during this first operation is:

$$L\ d \sin \alpha \quad \ldots\ldots (1)$$

where L is the length hp of the tracing arm from hinge to point. But the area of the rectangle $pqmn$ is:

$$A = d\ L \sin \alpha \quad \ldots\ldots (2)$$

and that is also the area of the type figure $pqrs$ as already indicated.

As for the other three operations: whatever the effect of swinging the arm from $h'q$ to hr may be, the swinging from hs to hp will neutralize it, while the third operation, with the tracing arm directly over the axis xx', will have no effect, for the axis of the wheel is then parallel to the line xx' and the wheel will slide without rolling. The reading of the wheel after the tracing is complete is consequently exactly the same as it was after the final operation, during which the point moved from p to q, and so far as the determination of area is concerned the tracing might just as well stop there.

*Pages 147, 211 and 248, April, May and June, 1900.

An extension of this method to a combination of several figures, like *pqrs* in Fig. 6, is apparent without going into detail. Again, starting at *w*, the figure *wuvr* may be traced, and then the figure *wrspq* and the difference of initial and final reading multiplied by the instrumental factor will give the sum of the areas of the two simple figures, or the area of *pqwuvsp*. Moreover, the tracing of *rw*, first up and then down, may be omitted and the contour traced

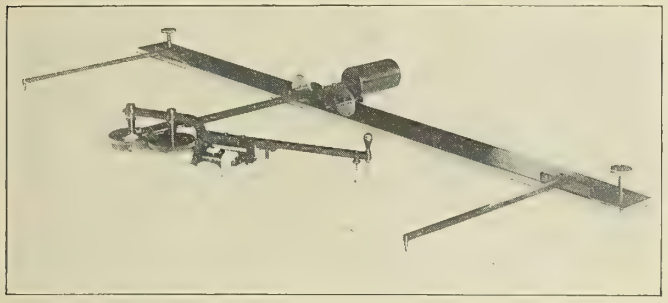


FIG. 1.

directly. In like manner any number of similar figures may be annexed until a complicated contour is measured, including areas like *rvdc* below the axis as well as areas above. To prove this last statement, it is sufficient to consider that in tracing the line *dc* in Fig. 7 the wheel rolls forward just as it does while the line *pq* is traced in Fig. 5; in either case, if one were looking along the arm from the hinge toward the wheel, the latter would be seen to turn in the direction of the hands of a watch.

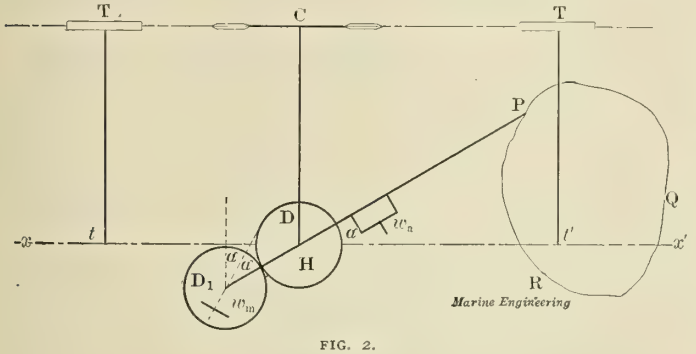


FIG. 2.

In all the preceding discussion it has been tacitly assumed that the figures, simple or complex, shall be traced in right-handed rotation; if any area is traced in the contrary direction its value will be subtracted instead of added.

To pass from the typical figure to any contour, for example to the irregular shape of Fig. 8, it is sufficient to consider that an approximation to the area may be obtained by tracing a contour made up of horizontal lines and circular arcs as indicated, and that we can make the approximation as close as desired by taking as many and as small steps as we please, so that at the limit we

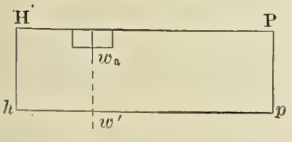


FIG. 3.

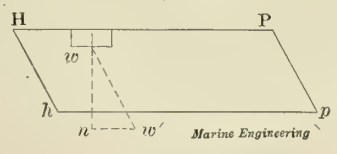


FIG. 4.

may get the correct area by tracing the contour of the figure itself. This conception is the fundamental conception of integration, and must be grasped if a correct idea of the action of a planimeter or integrator is to be obtained; it is thought that its statement, separate from and after the explanation of the mechanism of the instrument, will avoid confusion.

Before taking up the description of the method of determining moments by aid of the instrument represented by Fig. 1, it may be well to consider the nature of the property itself. The primary idea of a movement is simply the product of a force by its distance from an axis about which it may cause rotation. The weight of a body being a force, it is natural to speak of the moment of its weight with regard to an axis; the weight being assumed to act through the center of gravity. If we wish to find the moment of a system of bodies we find the moment of each separately, and add the several moments to find the total moment.

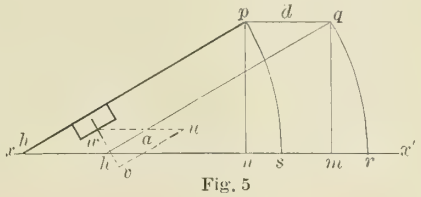


Fig. 5

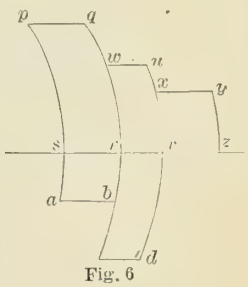


Fig. 6

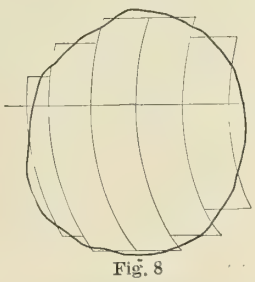


Fig. 8

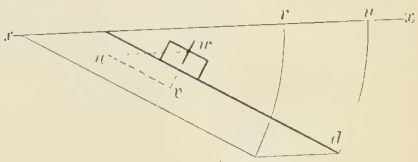


Fig. 7

A very natural extension of this idea gives for the definition of the moment of a plane figure like *ab* Fig. 9, with regard to the axis *xx'*, the product of the area of the figure and the distance of its center of figure from the axis. If the area is one square inch and if its center of figure is one inch from the axis, then the moment is unity. If in Fig. 9 *cd* is considered to represent four

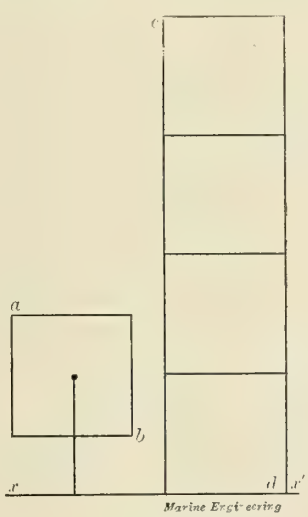


FIG. 9.

adjacent squares, each one inch on the side, then the moments of the several squares will be:

$$\tfrac{1}{2} \times 1,\ 1\tfrac{1}{2} \times 1,\ 2\tfrac{1}{2} \times 1,\ 3\tfrac{1}{2} \times 1,$$

and the sum or total moment will be 8. This result can be otherwise obtained by multiplying the area by half the height of *cd*; thus:

$$4 \times \tfrac{1}{2} \times 4 = 8.$$

A little consideration will show that the moment of any rectangle about its own base as an axis can be obtained by multiplying the area by half the height; so that if the height is represented by *h* and the breadth by *b* the moment will be:

$$M = \tfrac{1}{2} h \times area = \tfrac{1}{2} h \times h\, b = \tfrac{1}{2} h^2 b \ldots\ldots\ldots (3)$$

Turning to Fig. 5 it appears that the moment of the rectangle *pqmn* is equal to:

$$\tfrac{1}{2}\,\overline{pn}^2 \times pq.$$

But if the length of the tracing arm is represented by L and the angle *phn* by α, then *pn* may be replaced by $L \sin \alpha$; and of course we may represent *pq*, the distance the tracing point moves, by d. The moment of the rectangle is consequently:

$$\tfrac{1}{2}\,L^2 \sin^2 \alpha \times d,$$

or, changing the order:

$$M = \tfrac{1}{2}\,L^2\, d \sin^2 \alpha \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and since the typical figure can be made from the rectangle *pqmn* by taking away the figure *psn* and adding the equal figure *qvs*, the above expression represents the moment of the typical figure with reference to the axis xx'. If we can show that the instrument represented by Fig. 1 gives a record equivalent to the expression (4) it will be evident that it can be used to measure moments as well as areas.

To show that this can be done, let it first be noted that the guiding arm has rigidly fixed to it a screw-gear wheel and that the tracing arm carries an equal gear wheel engaging with it; this makes the familiar gear known as the sun and planet wheel, which has the property that the gear D_1, Fig. 2, on the tracing arm turns through twice the angle α of that arm. The gear D_1 carries a measuring wheel which is exactly like the area wheel w_a except that its axis is at right angles with the axis xx' when the tracing arm coincides with that axis. Consequently the axis of the moment wheel makes the angle $90^\circ - 2\alpha$ with the axis xx' when the tracing arm is at the angle α with the axis, as shown in Fig. 10. It is evident that when the tracing point P, Fig. 2, moves a distance d on a line parallel to the axis xx' the wheel will roll and slide a like distance, and that it will roll a distance:

$$hi = d \sin\,(90^\circ - 2\alpha) = d \cos 2\alpha \ldots\ldots\ldots (5)$$

The scale on the wheel w_m is so graduated that it appears to turn backwards; that is, the reading diminishes during this operation, and consequently a negative sign is given to the quantity in equation (5).

When the line *sr* of the typical figure *pqrs* of Fig. 5 is traced the tracing arm coincides with the axis xx', and the moment wheel w_m, whose axis is then perpendicular to the axis xx', rolls the entire distance d, and the reading of the wheel increases during this operation. As was found to be the case with the area wheel in the discussion of area, it can be readily seen that whatever effect is produced on the moment wheel w_m by tracing the arc *qr* will be counterbalanced by tracing the arc *sp*. So that the net effect of tracing the typical figure *pqrs* will be equal to:

$$d\,(1 - \cos 2\alpha) \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

But a well-known trigonometrical equation allows us to replace the expression (6) by the expression:

$$2\,d \sin^2 \alpha \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

If this expression is introduced into the equation (4) already deduced for the moment of the typical figure, we must write

$$M = \tfrac{1}{4}\,L^2\,(2d \sin^2 \alpha) = \tfrac{1}{4}\,L^2\, d\,(1 - \cos \alpha) \ldots\ldots (8)$$

The extension of the operation of measuring moments by aid of the integrator to an area like Fig. 6 and to any contour like Fig. 8 is in general similar to that already given for areas. There is, however, one important difference, for the instrument subtracts moments of figures or parts of figures that lie below the axis xx'. To see that this is true, consider that in Fig. 10, if one looks at the side of the wheel W_m, to which the scale is attached, it will appear to turn to the right, like the hands of a watch, while in Fig. 11 it turns in the contrary direction.

Coming now to the moment of inertia, we may start with the idea that this property differs from the moment in that the multiplier is to be the square of the distance instead of the distance itself; a statement that is purposely left incomplete, for the time being, because the proper conception of moments of inertia requires the use of the calculus or its equivalent. An attempt will be made to build up the conception from Fig. 9 in somewhat the same method as was used for moments. Let us first multiply each square inch by its distance from the axis xx' and sum up, giving

$$(\tfrac{1}{2})^2 \times 1 + (\tfrac{3}{2})^2 \times 1 + (\tfrac{5}{2})^2 \times 1 + (\tfrac{7}{2})^2 \times 1 = 21,$$

as a first approximation. For a second approximation let us take half squares getting

$$(\tfrac{1}{4})^2 \times \tfrac{1}{2} + (\tfrac{3}{4})^2 \times \tfrac{1}{2} + (\tfrac{5}{4})^2 \times \tfrac{1}{2} + (\tfrac{7}{4})^2 \times \tfrac{1}{2} + (\tfrac{9}{4})^2 \times \tfrac{1}{2} + (\tfrac{11}{4})^2 \times \tfrac{1}{2} + (\tfrac{13}{4})^2 \times \tfrac{1}{2} + (\tfrac{15}{4})^2 \times \tfrac{1}{2} = 21\tfrac{1}{4}.$$

If the same process were extended to quarter squares the result would be 21 10-32; and a continuation of the method would show that as the width of the elementary area decreased the result would approach the limit 21 1-3. And this result can be obtained by taking one-third the cube of the height of the figure; that is

$$\tfrac{1}{3} \times 4^3 = 21\tfrac{1}{3}.$$

If this conception be united with the very simple idea that the moment of inertia increases directly with the *width* of the figure we may write down the common form for the moment of inertia of a rectangle about an axis at its base,

$$\tfrac{1}{3}\,bh^3,$$

when b is the base and h the altitude. It will be appreciated that the dimension just given is rather an illustration than a demonstration, and that it is left incomplete to avoid prolixity.

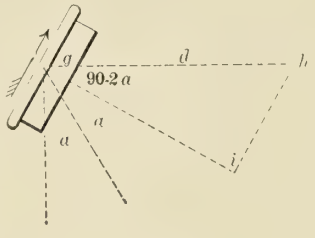


FIG. 10.

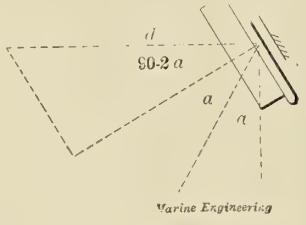


FIG. 11.

The instrument which is designed to measure moments of inertia is represented by Fig. 12, in which the tracing arm carries two circular rack or half-gears, which engage two equal pinions, each of which carries a measuring wheel, making three measuring wheels in all; namely, the area wheel, the moment wheel, and the moment of inertia wheel. The area wheel is carried by the tracing arm in the usual way, and the moment wheel is carried by a pinion geared at a ratio of two to one to the tracing arm; while the moment of inertia wheel is carried by a pinion that is geared to the tracing arm with a ratio of three to one and consequently its axis is turned through three times the angle to which the tracing arm is set, and this is the essential feature of the measurement of moments of inertia.

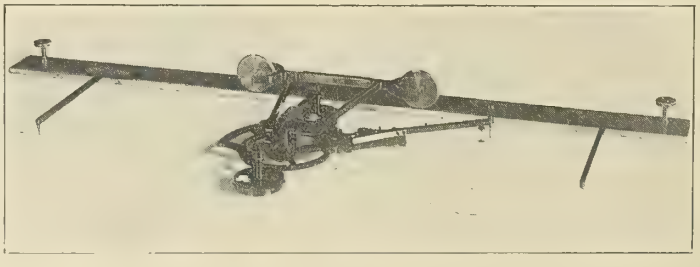


FIG. 12.

Turning to the type area, *pqrs*, Fig. 5, it may be admitted that its moment of inertia is the same as that of the rectangle *pqmn*; that is, to

$$\tfrac{1}{3}\,\overline{pq}\,\overline{pn}^3.$$

But if L is the length of the tracing arm, and α is the angle between it and the axis, and if the line pq is represented by d, then the moment of inertia of the typical figure may be written:

$$\tfrac{1}{3}\, d\, (L \sin \alpha)^3,$$

or

$$I = \tfrac{1}{3}\, L^3\, d \sin^3 \alpha.$$

But the cube of the sine of an angle may be expressed as follows:

$$\sin^3 \alpha = \tfrac{3}{4} \sin \alpha - \tfrac{1}{4} \sin 3\,\alpha.$$

so that

$$I = \tfrac{1}{4}\, L^3\, d \sin \alpha - \tfrac{1}{12}\, L^3\, d \sin 3\,\alpha \ldots\ldots\ldots\ldots (9)$$

and it becomes necessary to show how the instrument may be made to give the corresponding records.

As has already been pointed out, the moment of inertia wheel has its axis turned through three times the angle which the tracing arm makes with the axis xx'; and now attention must be called to the fact that, like the area wheel, its axis is perpendicular to the axis xx' when the tracing point is on that line. Though this relation cannot be determined from inspection of Fig. 12 it is easily understood, as are also the consequences of that arrangement. Thus in tracing the typical area, $pqrs$, Fig. 5, the portion pq is the only one that need be considered, for the wheel slides without rolling when rs is traced, and the effect of tracing the arc qr is compensated by tracing the arc sp. While the line pq is traced, the moment of inertia wheel slides and rolls the distance d, but its axis makes the angle $3\,\alpha$ with that line, and consequently it rolls the distance $d \sin 3\,\alpha$, a factor which appears in equation (9); the other term of that equation contains the factor $d \sin \alpha$ that has already been found to apply to the area wheel, and consequently the instrument furnishes all that is required for using that equation. It should be pointed out that the moment of inertia wheel turns forward when the line cd, Fig. 7, is traced, following the method of the area wheel and for the same reasons; and that consequently it *adds* the moments of inertia of figures or parts of figures which are below the axis xx', as it should, for though a distance below the axis may be considered to be negative, its square will be positive.

The tracing arms of the integrators represented by Figs. 1 and 12 are eight inches long, measured from the hinge to the tracing point, and the perimeters of the area wheel and the moment wheel are 2.5 inches (diameter 0.7957 inch); the moment of inertia wheel, as will be explained later, differs slightly. Now it has been shown that for the typical figure the area is

$$L\, d \sin \alpha,$$

where d is the distance traced parallel to the axis xx' and α is the angle which the arm makes with that axis. If the distance $d \sin \alpha$ which the wheel rolls is 2.5 inches, the wheel will make one complete turn and the area recorded will be

$$8 \times 2.5 = 20 \text{ square inches};$$

and this is the factor by which the difference of reading of the area wheel is to be multiplied to find the area of the figure in square inches.

In like manner, if the net effect of tracing the typical figure is one turn of the wheel, so that

$$d\, (1 - \cos 2\,\alpha) = 2.5 \text{ inches},$$

then the moment is

$$\tfrac{1}{4}\, L^2\, d\, (1 - \cos 2\,\alpha) = \tfrac{1}{4} \times 8^2 \times 2.5 = 40 \text{ square inches};$$

which is the factor for moments.

Coming now to the moment of inertia, it is to be noted that the wheel is a little smaller than the area and the moment wheel, in fact its perimeter is 2.3438 inches, a diminution which is chosen to get a convenient factor for computation.

In equation (9)

$$I = \tfrac{1}{4}\, L^3\, d \sin \alpha - \tfrac{1}{12}\, L^3\, d \sin 3\,\alpha.$$

If the area wheel makes one complete turn the product of the distance rolled by one-fourth of the arm cubed gives

$$\tfrac{1}{4} \times 8^3 \times 2.5 = 320.$$

On the other hand, the factor for the moment of inertia wheel is

$$\tfrac{1}{12} \times 8^3 \times 2.3438 = 100.$$

Consequently we have for the integrator with an eight-inch tracing arm the following equations:

$$\text{Area} = 20\,(a_2 - a_1),$$
$$\text{Moment} = 40\,(m_2 - m_1),$$
$$\text{Moment of inertia} = 320\,(a_2 - a_1) - 100\,(i_2 - i_1);$$

where a_1 and a_2 are the initial and final readings of the area wheel, m_1 and m_2 are the readings of the moment wheel and i_1 and i_2 those of the moment of inertia wheel.

The instruments represented by Figs. 1 and 12 have a second tracing point at half the length of the arm, that is, at four inches from the hinge, which point may be used for tracing small figures. With a four-inch tracing arm the factors are:

$$\text{Area} = 10\,(a_2 - a_1),$$
$$\text{Moment} = 10\,(m_2 - m_1)$$
$$\text{Moment of inertia} = 40\,(a_2 - a_1) - 12.5\,(i_2 - i_1).$$

These instruments, when in good condition, as they come from the makers, are very accurate and can be relied upon to give good results for area and moments; unfortunately they are much less satisfactory for moments of inertia because the formula involves taking the difference of two numbers which are liable to be not very unequal, and this relatively small difference is affected by the

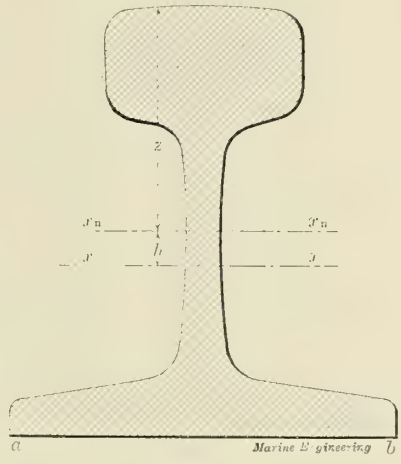


FIG. 13.

entire instrumental error. If we take, for example, a circle tangent to the axis xx', we may have the following readings:

	Area	Moment	Moment of Inertia.
First	1.832	4.495	3.721
Second	2.460	5.123	5.103
Difference	0.628	0.628	1.382

$$\text{Area} = 20 \times 0.628 = 12.56,$$
$$\text{Moment} = 40 \times 0.628 = 25.12,$$
$$\text{Moment of inertia} = 320 \times 0.628 - 100 \times 1.382,$$
$$= 200.96 - 138.2 = 62.76.$$

Now the vernier can be read by estimation to half a thousandth, so that the area may be affected by an instrumental error of one unit in the last place; roughly the error may be taken to be one in the thousandth. The moment should be assigned as large an error, that is, an error of two units, in the second decimal place. The same proportionate error should be attributed to both wheels that affect the moment of inertia; in the example the error should be 0.2 in 200.96 and something larger than 0.1 in 138.2, so that the final error is likely to be at least 0.2 in 62.8, that is about one in three hundred. Even with much practice and skill readings as exact as those quoted are seldom obtained, and practical errors are likely to be much larger.

An important use of this instrument is to determine the moment of inertia of a section like Fig. 13, about an axis through its

center of gravity. For this purpose an axis like xx may be taken at random and the area, moment, and moment of inertia can be measured. The quotient obtained by dividing the moment by the area will give the distance h of the center of gravity from xx. To find the moment of inertia about the axis $x_n x_n$, subtract from the moment of inertia already determined with reference to the trial axis xx, the product obtained by multiplying the area by the square of the distance between the axes.

$$I_{xn} = I_x - Ah^2.$$

Should it be desired to use these instruments on a drawing to a reduced scale, we may proceed as with full sized drawings, and then multiply the area in square inches by the square of the reciprocal of the scale, multiply the moment by the cube of that quantity and the moment of inertia by the fourth power. For example, a drawing to the scale of three inches to the foot is to quarter scale. The area is to be multiplied by 16, the moment by 64, and the moment of inertia by 256.

Finally it may be convenient to write down the usual form for the properties which we have been discussing, in the notation of the calculus.

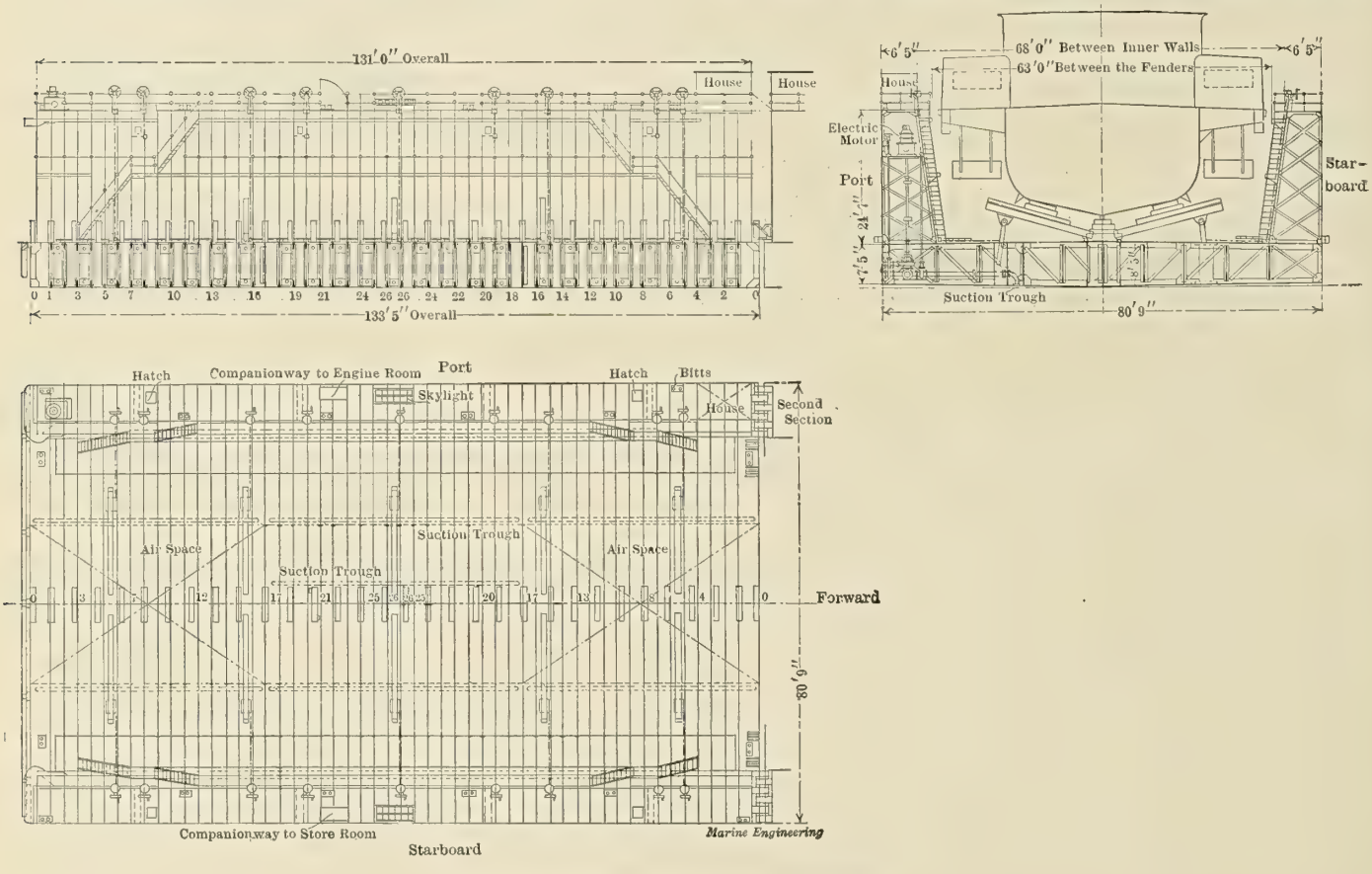
$$\text{Area} = \iint dx\,dy = \int y\,dx,$$
$$\text{Moment} = \iint y\,dx\,dy = \tfrac{1}{2}\int y^2\,dx,$$
$$\text{Moment of inertia} = \iint y^2\,dx\,dy = \tfrac{1}{3}\int y^3\,dx.$$

THE FLOATING DOCK OF THE AKTIEN GESELLSCHAFT "NEPTUN," SHIPYARD AND ENGINE WORKS, OF ROSTOCK, IN MECKLENBURG.*

BY KARL ZUBLIN.

A floating dock, which, on account of the new principle involved in it, is especially noteworthy, was set to work last summer by the above mentioned shipyard. Prior to that time the "Neptun" shipyard had not been in possession of docking appliances of their own. For laying vessels dry recourse had hitherto been had to a slipway, which sufficed for vessels of at most 800 tons in weight, with length of about 250 feet, and breadth of about 36 feet. Since, however, the yard had latterly dealt with larger work and even built vessels of about 3,500 tons equipped weight and 370 feet length, such as the *Prinz Sigismund*, for the Hamburg-American Line, it had become more and more necessary for the company to have docking accommodations of their own for vessels of this larger class. The completion of the arrangements for the establishment of a ferry service between Warnemünde and Gedser gave the last impulse to the scheme for the building of a new floating dock.

This dock is constructed on the new "Dieckhoff" system (German patent, No. 150,572; American patent No. 755,854, of March



PLAN AND SECTION OF NEPTUN FLOATING DOCK.

The Mitsu Bishi Dock Yard and Engine Works, of Nagasaki, Japan, are to build two 15,000-ton steamships for the Toyo Kisen Kaisha's service from San Francisco to the Orient. The other Japanese steamship company, Nippon Yusen Kaisha, is having built at the same place a 6,000-ton steamer for its Seattle service. Material for these new steamships and six others is being bought in the United States, and shipments are now being made from San Francisco. Locomotives, rails, and raw material for the new water works at Yokohama are also being shipped, the purchases having been made in this country partly on account of the prices and in part due to the speedy delivery.

29, 1904). The calculations and design of the hull of the dock, as also of its machinery and outfit, were the work of Professor Dieckhoff, of Charlottenburg, while the construction was undertaken by the "Neptun" Shipyard itself.

The dock, which is illustrated herewith, is of the usual U-shaped cross section, and consists in the longitudinal direction of two separate sections which are exactly symmetrical. In this arrangement provision is made for an eventual increase in length, and consequently also of the carrying capacity of the whole structure, by the addition of a third section. The object of dividing the

*Translated for MARINE ENGINEERING.

whole structure into several separate parts was that in the case of the docking of several small vessels at once each section might be able to work independently of the others. In this way the work on a vessel which has already been raised by one of the sections can always proceed unhindered while another section is docking or undocking a second vessel. The clear width of the dock has been made comparatively great in view of the eventual docking of the broad paddle steamers plying between Warnemünde and Gedser. In the drawing, a single section is shown, and in the photographs, two such coupled together, viewed from the shore.

END VIEW OF DOCK, SHOWING APPROACHES.

The dimensions are as follows:

	m.	ft.
Length over the pontoon of one section	40.662	133.8
Length over the pontoons of the two existing sections	81.650	268.
Length of each side wall	39.91	131.
Extreme breadth over hull of dock	24.612	80.8
Inner width between side walls at top of dock	20.7	67.9
Inner width between wooden fenders at top of dock	19.2	63.
Width of each side wall at top of dock	1.956	6.42
Width of each side wall at height of dock floor	3.15	10.33
Height of pontoons at centre line of dock	2.562	8.41
Height of pontoons at side of dock	2.262	7.42
Height of side walls	7.5	24.6
Height of keel blocks	1.2	3.94
Greatest draught of vessels to be raised	5.5	18.
Freeboard of the side walls therewith	0.65	2.13
Carrying capacity of the two existing sections when the pontoons are allowed suitable freeboard	3,000 tons	

Each pontoon is provided with three longitudinal and two transverse bulkheads. With the exception of the central division all these are made watertight, so that nine separate watertight compartments thus result. The two middle-line end compartments form the necessary air compartments, the rest being arranged for the reception of water.

The transverse frames take the form of lattice girders. As may be seen in the longitudinal section, successive pairs of transverse frames are connected together at certain points above and below, so as to form a double girder. This arrangement takes the place of the usual diagonally arranged longitudinal structural parts, and has the advantage, that passage-ways are provided between the successive double girders, as also completely unencumbered spaces for the laying of the piping. In the general arrangement of the transverse frames special care must be taken, that on the floors of the compartments, the water, and under their top-plating, the air, can circulate unhindered within the various frame spaces. In view of this both bottom and top plating are laid at a suitable angle with the horizontal on each side. In the case of the deck over the bottom compartments, this also carries with it the advantage, that the water above it can run off quickly, leaving a dry floor for work under the bottoms of the vessels. Under the dock floor a strong longitudinal girder is arranged within a short distance of the central girder. An advantage connected with this arrangement is, that the keel blocks are not obliged to be placed at one particular point above the transverse frame, but can, in case of repairs to a vessel, be arranged wherever desired, without incurring the danger of bulging in the pontoon top-plating. In accordance with practical requirements, the outer walls of the pontoon are made especially strong.

Each side wall contains in its upper portion a watertight compartment, serving at the one side of the dock as an engine room, and at the other as a store room. These spaces at the same time ensure the unsinkability of the whole dock in case the water valves should, through carelessness, be left open. Their size, indeed, is so great that a portion of each of them also serves as an air chamber, and together with the air chambers in the bottom pontoon are capable of supporting the equipped weight of the dock. For ensuring the necessary longitudinal stability of each separate dock-section, the total water space of each side wall is divided by means of two watertight bulkheads into three separate water-chambers.

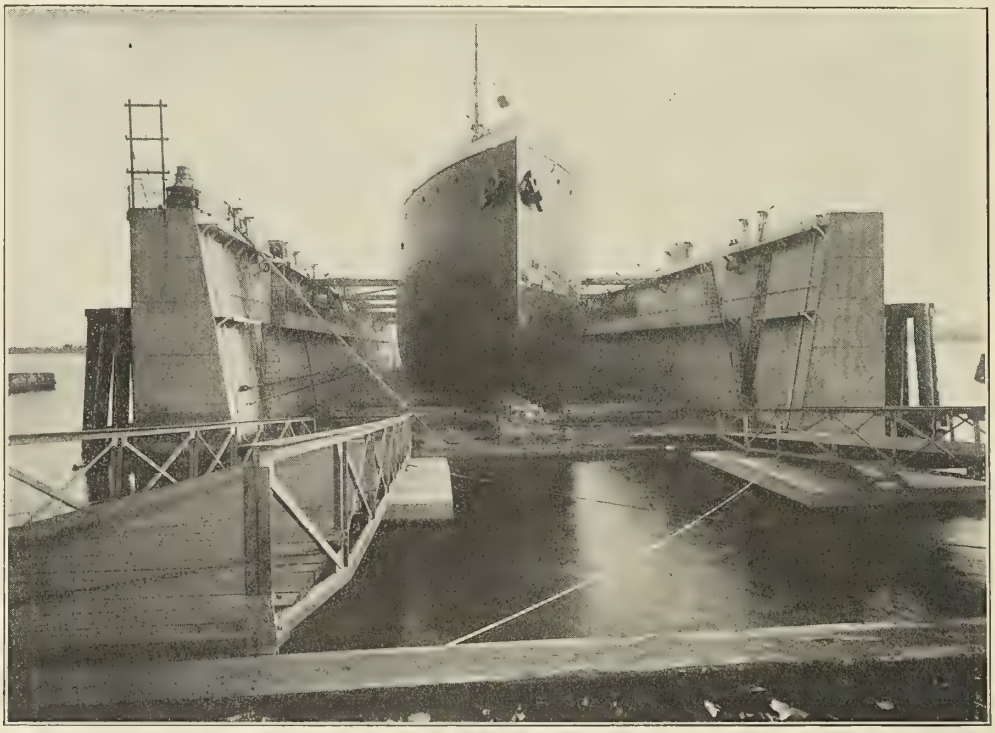
The equipment of the dock is of very considerable extent. The keel-blocks are made relatively high in order to allow of convenient working under the bottom of a vessel, and are kept very broad and close together in order to minimize the bearing pressure. They consist, in each case, of four open baulks, piled one upon another. Below these are pairs of tapered pieces, which are arranged for wedging up, and fitted with driving rings. The various parts

are suitably secured to prevent their swimming away or becoming displaced.

The bilge supports consist in each case of two beams, arranged one above another in such manner that the upper one is free to bend to a certain extent, according to the load that may bear on it. The lower beam is stepped in a socket upon the dock floor not far from the central girder, and is so arranged that its outer end can be moved up and down by means of a rack and pinion. The pinion is actuated from the deck of the side wall by means of shafting and toothed and worm wheels with an automatic braking arrangement. The racks work in shafts extending watertight from the top to the bottom of the pontoon. This whole arrangement is not a very simple one, but it has some noteworthy advantages. By its adoption the building up of the bilge blocks to suit the rise of floor of the docked vessel becomes unnecessary, and vessels of the most varied kinds can be shored up without the forms of their bottoms having to be taken into consideration beforehand. Furthermore, this arrangement enables the bilge supports to be easily loosened and fixed at any time.

made of strong wooden planking, is fitted, along which, at the same time, the heavy ship plates may be transported on bogeys. Bollards, fairheaders, companions, skylights, etc., are suited to practical requirements.

Each section of the dock has a dockmaster's cabin placed on one of the side decks at the end adjoining the next section. At the opposite end is fixed an electrically worked capstan with vertical drum, which is used for drawing the vessels into or out of the dock. Along the outer longitudinal and end walls continuous wooden fenders are fixed to short vertical Z-irons in such a manner that they do not come directly into contact with the skin of the dock, and thus do not afford opportunity for the dreaded rusting which is apt to be produced by contact between wood and iron. The fenders are arranged above the pontoon deck so as to reduce the danger of the sinking of the dock. In case these be heavily strained, the wall plating of the pontoon will, under this arrangement, not be liable to suffer damage. The outer ends of the pontoon are provided each with five vertical wooden fenders. At the entrance end the end-plating of the side



DOCK WITH A SHIP IN POSITION.

The side-shores are placed in the upper part of the side wall on each side, and arranged to slide on rollers in watertight shafts. These also are worked from the deck of the side wall by means of shafting and toothed and worm wheels. Inner ends of some of the side-shores are provided with rollers, so that in the docking of a vessel, the latter can conveniently be guided, moved, or heeled.

Along each of the inner side walls and extending as far as possible out the entire length of the dock, two dock-shelves made of corrugated iron are arranged, which serve as gangways or as supports for stagings and side-shores in cases of vessels undergoing repairs. The upper dock-shelf lies at a lower level than the side deck, so as here also to provide a good point of support for side-shoring of docked vessels. The inner edge of the dock-shelf is in part provided with wooden fenders. For the lower dock-shelf a handrail is fixed at suitable height along the inner side wall. Several easy stairs lead from the bottom pontoon to the side decks. The lower parts of the former are made to fold, so that in case of need the whole of the clear width between the side walls may, up to a certain height, be made available. Above the floor of the bottom pontoon a broad gangway, walls is in each case provided at the height of the upper dock-shelf with a strong wooden fender as a protection against drifting vessels.

The connection between the sections of the dock is in the form of a hinge arrangement. If the sections of the dock have the tendency to lean toward each other in the longitudinal direction, motion can take place unhindered about the middle of the hinge pin, to a very considerable extent, so that the bulging in of the neighboring end plating which so often takes place in docks of this kind is avoided. For the case in which the sections of the dock tend to lean in different transverse directions only a small amount of play is provided, so that the separate parts can afford support to each other. This support is provided by strong brackets fitted on the pontoon which are made to fit into each other in such a manner that no considerable transverse movement of the sections, one with another, can take place. The whole coupling arrangement is such as to admit of simple and rapid connecting and disconnecting, which is necessary when a single section may be called upon to work independently. It also makes it possible to couple together two sections with a considerable interval (in this case

about 30 feet in length) between them, so that a relatively long vessel can be docked, the middle part of which is not then supported from the floor of the dock, but hangs free.

In regard to pumps, each section has two centrifugal pumps with vertical shafts. In order to reduce the height from which the pumps have to draw, and to enable these during ordinary work to reach the point of inlet of the water, they are arranged as low down in the dock as possible. They are fitted directly over the main line of piping, which, resting on the floor plates, runs along one side of the dock. From the main pipe to the various water chambers of the bottom pontoon are fitted branches, which can be shut off by means of water-valves. These branch pipes are connected with a trough which runs along the bottom of the pontoon over the whole length of the various water chambers. In the bottom plates holes are cut above the troughs so that these latter are directly connected with the spaces between the several floor plates. The trough arrangement enables each dock section to be completely emptied by the main centrifugal pumps. It avoids the difficulty otherwise met with that water lodging between the floor plates must either be removed by special piston pumps or remain as irremovable ballast, in which latter case the carrying capacity of the dock is reduced.

Each centrifugal pump is directly coupled to a vertical electric motor fixed in the engine room. There are thus two completely independent sets of pumping machinery provided for each dock section, so that, in case one of these should break down, the dock can be pumped out by the other one alone. The pontoon-chamber inlets and outlets through the skin of the dock are provided with suitable sieves and non-return flaps. Each water chamber of the side walls can be shut off by a simple flap-valve. The total capacity of the pumping plant is so adjusted that, according to the weight of the vessel to be raised by the dock, the time required for emptying the latter is 35 minutes for a weight of about 1,000 tons and 80 minutes for a weight of about 3,000 tons. Each motor is of 20 horsepower, so that the whole plant necessary for the two existing sections has an aggregate of 80 horsepower. The pumping plant which has to be provided in connection with the Dieckhoff patent is thus relatively very small, and its working efficiency is relatively very large. This good result is due on the one hand to the small variation in the depth from which the pumps have to draw; and on the other to the circumstance that instead of the pump suctions having to be throttled, during the process of filling, by manipulations of the check-valves, the suction valves, after the vessel has been berthed, remain completely open, the horizontal adjustment of the rising dock being effected entirely by the help of the check-valve of the side wall.

The lowering of the empty dock from the position in which the pontoon floats with its normal freeboard, is accomplished on the average in 40 minutes. The lowering of the dock with a vessel resting on it varies with the collective weight, but is accomplished in a much shorter time.

The manipulation of the check-valves for the water chambers of the pontoon, and of the side walls, is effected from the dockmaster's cabin as a central station. The latter further contains level indicators for the longitudinal and transverse directions respectively. The whole dock is manœuvred from the central station referred to and speaking-tube connection between the latter and the engine room is provided. After a vessel has been berthed one man is able to fill the dock without having to trouble himself with what may be taking place outside of the dockmaster's cabin. To enable the height of the water in the pontoon chambers to be observed automatic indicators are provided. To enable water to be thrown upon docked vessels for fire quenching and similar purposes, there is a special electrically-driven pump with suitable piping running along the dock, provided with the necessary branches and hose couplings. To enable tools worked by compressed air to be used in the dock a pipe is laid throughout its length provided with branches, to which the compressed air can be led from the shore.

In the engine room is installed a small steam boiler which in winter supplies steam for warming purposes and for thawing the valves and working parts. The lighting of the whole dock is electric. The inner lighting is done by incandescent lamps; the outer by arc lamps. The latter are suspended on booms that can be swung out, or, if necessary, lowered to the dock floor, so that light may be thrown under a vessel's bottom.

The position of the dock in the water is such that its longitudinal axis lies perpendicular to the bank. The traffic with the land is carried on, as shown in the photographs, by means of two broad gangways which at the ends resting on the quay are so arranged that they can turn, and at the opposite ends are borne by suitable pontoons. To preserve its longitudinal direction the dock is guided in the usual manner between several groups of strong piles.

The first floating dock built on this principle was set to work by the firm of H. C. Stülcken Sohn, of Hamburg, on the 23d of April, 1902. In its case both the calculations and the practical comparative trials have established the fact that, compared with the ordinary system as applied under like conditions—that is to say, with given carrying capacity, stability, and time for emptying—the following savings are effected: 47 percent saving in size of pumps and piping; 38 percent saving in size of power plant; 38 percent saving continuous mean consumption of power. Meanwhile, the further advantage is here obtained, that the consumption of power is nearly proportional to the weight of the vessel to be raised, which in a dock on the ordinary system is far from being the case.

Another practical comparative trial with the Stülcken dock, which for this purpose was with very little difficulty arranged to work temporarily on the ordinary system, gave the following result: Under the same conditions—that is to say, with like displacement and draft of the docked vessel, and with the like performance of the power plant—the time required for pumping was 50 minutes when the dock was worked on the ordinary system, and 30 minutes when it was being worked in the Dieckhoff system. This trial, then, showed a saving of 40 percent in time and consumption of power.

The new floating dock of the "Neptun" shipyard has been almost continuously occupied since the beginning of last summer, and has shown itself efficient in every respect.

HOW TO INDICATE THE ENGINE.

BY GEORGE P. PEARCE.

This short article is intended to help the beginner to take his first cards, and so get him interested enough to make a study of the subject; he will then be able to set valves, instantly find out where various troubles lie, run his engine at the highest economy, and thus get the best results.

THE INDICATOR.

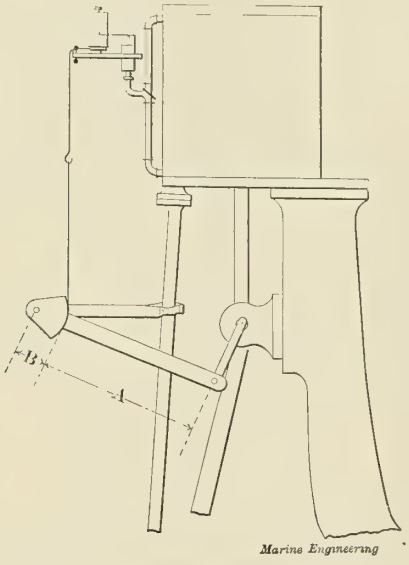
The first thing to do is to thoroughly clean the indicator, and see that all parts are in good working order. Great care should be exercised in handling, as it is very easy to bend the parallel motion or otherwise destroy its accuracy. Next select the spring; this should be of such strength that it will make the diagram about 1 3-4 inches high, unless the engine is a very high-speed one, when it should be only about 1 1-4 inches. A general rule for getting the right spring is to divide the boiler pressure in pounds per square inch by the desired height of the card in inches, and the result will be the number of the spring. I might mention here that the number of the spring means the steam pressure in pounds per square inch which will be required to lift the indicator pencil one inch; or, a No. 40 spring used under 40 pounds pressure would make a diagram one inch high.

Having selected the right spring, attach it to the piston, carefully oil and get everything ready, place a card on the drum, and get someone to revolve it slightly while you lightly press the pencil against it and draw a line. Now, if the engine is running condensing, measure the distance in inches from this line to

the bottom of the card and multiply it by the number of the spring; the quotient should be at least 25 in order to insure plenty of room for the vacuum line. The distance from the line to the top of the card should be about 2 inches. If these are not right the indicator pencil must be adjusted until you get it in the right position; for adjusting the pencil you will find instructions in the box; this arrangement varies with different makes of indicator. After this the indicator is ready to attach to the cylinder.

THE ENGINE CYLINDER.

Next see if the cylinder has indicator pipes for attaching the indicator. If it is an old engine which has never been indicated, you will probably have to drill into the steam passages yourself. To do this it will be necessary to carefully measure how far the piston reaches at each end, and then drill into the clearance spaces, or get as much of the holes as possible there, and chip it into a slot shape so as to be sure that it is of ample size, and that no wire drawing of the steam can take place. The holes should be for a 1-2-inch pipe tap and should be carefully tapped. Next, two short pieces of 1-2-inch pipe with elbows on, should be



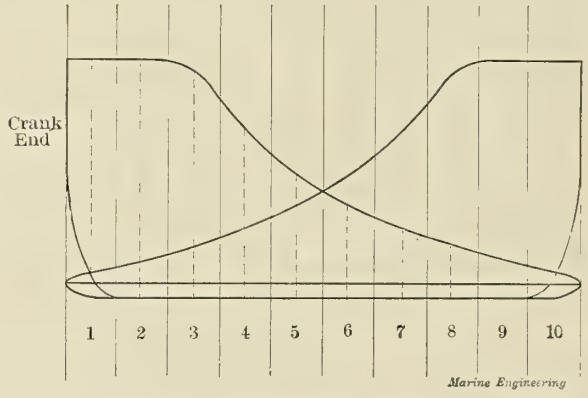
REDUCING MOTION.

screwed in, so as to make a steam-tight joint. I want to warn you against using red lead or any kind of paste in making these joints, because of the bits which are apt to become loosened and get into the indicator cylinder and thus ruin it. If there is any steam leakage, the joint should be loosened and some cotton or twine wound around the threads; this will make them steam tight. Next, two pieces of pipe must be screwed into the elbows, the two ends meeting in the special three-way valve, which is furnished with the indicator. This should be fitted so that the valve will be about in the center of the cylinder. Now, give these pipes a thorough blowing out in order to clean out all dirt, and then all is ready to connect the indicator.

As the indicator diagram should be only about 4 inches long it will be necessary to arrange some device which will reduce the motion of the crosshead to this amount. All modern indicators have a reducing motion attached; if yours is one of these, it will be necessary merely to put a 1-2-inch tap hole in the crosshead, thread the end of a 1-2-inch iron bar, and fit it in. This must be long enough to reach the indicator cord when coming straight from the reducing wheel, care being taken that the cord has a straight to and fro pull without any "wobble," as this would distort the card. If you have no reducing motion you will have to devise one. Perhaps the simplest, and at the same time a very accurate one to make, is the one shown in the cut. The proportions for the parts are: as the stroke of the engine in inches is to the length of the diagram (4 inches), so the length *A* should be to the radius *B*. It is all made from wood except the pins, and the general construction can easily be understood from the cut. The reason for using a quadrant for the cord to run on is to cause the cord to run always in a straight line, and so obtain correct cards.

INDICATING.

Now, attach the indicator; also attach the reducing wheel to the indicator. Next take the cord from the reducing wheel (I am speaking of the modern reducing motion, or wheel, which is attached to the indicator), and move the end or hook over the same distance which it will travel when attached to the iron rod in the crosshead, and thus assure yourself that it is of the right length and will not pull the indicator against the stop. Then throw the indicator out of gear, and by quickly moving the cord to and fro with the crosshead you will be able to catch the loop on the rod, and thus hook up the indicator into gear. Open the indicator valve and let the indicator operate for a short time, so that all will get warmed up. Then put a card carefully around the indicator drum, bringing the ends together just over



INDICATOR CARD PREPARED FOR COMPUTATION.

the prongs, or fork-shaped pieces of metal, and with a sliding motion slip it downwards until it reaches the end of the prongs; if done properly the paper will be quite tight upon the drum. Now, turn the three-way valve so that it will be closed to the back end of the cylinder and open to the crank end, and while the pencil is still oscillating, but not touching the paper, mark down the time in your note book; as the indicator is now thoroughly warmed up and its relative parts expanded, you can close the indicator cock, observing that the little bye-pass is open to the atmosphere. Then draw the atmospheric line by lightly pressing the pencil against the card for a few revolutions. Next open the indicator valve and lightly press the pencil once more against the card, holding it there for about a dozen revolutions of the engine; then disconnect the indicator, take off the card, and mark it "Crank end Number 1." The next card should be taken at once, but on this one get the atmospheric line, crank end, and back end without disconnecting the cord. Take it off, and upon comparing it with the first one you will be able to see which is the crank end. This you should mark, and you will be able to see if the valves are set about right. If everything is correct you should take the speed of the engine, mark down all data in your note book, and then take a card every fifteen minutes,* always remembering to get atmospheric line, crank end, and back end of engine, on each card; also keep filling in your notes. Remember that by getting two diagrams *quickly* on one card you minimize the risk of having the governor move between the first and second diagrams, which would make it appear as if the valves were set wrong.

*In some cases cards are taken every ten minutes.

CALCULATING THE INDICATED HORSEPOWER.

The easiest way to find the mean effective pressure is with a planimeter, but if you have not one, take one of the cards and carefully erect a perpendicular at each end of the atmospheric line, just touching the ends of the diagram. Divide this space into ten equal spaces, and draw ten lines, also at right angles to the atmospheric line. Now, take a point on the top of crank-end diagram line, just half way between the first two lines, and measure the length to a point between the same two lines on the bottom crank-end diagram line. Repeat this on each of the ten spaces (see cut), then add all these measurements together and divide by ten; this will be the average height. An easy way to do this is to take a strip of paper and mark off the first length on it, then mark off the second length, starting from the end of the first, and so on for the ten lengths. It then remains only to measure the length from the first point to the last, and divide by 10. Now, multiply this average length in inches by the scale of the spring and this will give you the mean effective pressure, or the average pressure that was acting on the piston during the stroke. This must be multiplied by the distance which the piston traveled in one minute under pressure from the crank end. This is equal to the number of revolutions multiplied by the stroke in feet. The product will give you the foot-pounds of work done on each square inch of the piston. Next, multiply this by the number of square inches in the area of the piston minus the area of the rod. This will give the total number of foot-pounds of energy developed per minute; divide this by 33,000, and the quotient is the indicated horsepower for the crank end of the cylinder. Repeat this on the back end diagram; but this time you will take the full area of the piston, because there is no rod on that end, unless it is an air compressor or tandem engine, when you will take out the rod area as usual. This result should about equal that for the crank if the valves are set correctly. Add the two together and the result will be the indicated horsepower of that cylinder, and the indicated horsepower of the engine, if it has only a single cylinder. Obtaining the indicated horsepower is only one of the many uses of the indicator, but it is the most usual one.

Launch of Turbine Steamer Dieppe.

On the 6th of April there was launched by the Fairfield Shipbuilding and Engineering Company, Limited, of Glasgow, the triple-screw turbine steamer *Dieppe,* which they have built for the London, Brighton, and South Coast Railway Company's passenger service between Newhaven and Dieppe. The steamer is 274 feet long, 34 feet 8 inches in breadth, and 14 feet 6 inches in depth, and is built to Lloyd's requirements for Channel purposes. She has a promenade deck, all fore-and-aft; also a shade deck of sufficient length to carry all the boats, extending right across the ship, and giving shelter to passengers on promenade deck. On the promenade deck there are the first-class deck saloon, first-class entrance, ladies' boudoir, and a number of special state rooms, all fitted up in a luxurious manner, and having vertical motion square windows. A special feature of the ship is the ventilation arrangements, electric exhausting fans being fitted in the entrance and saloons. A complete installation of electric lights will be fitted.

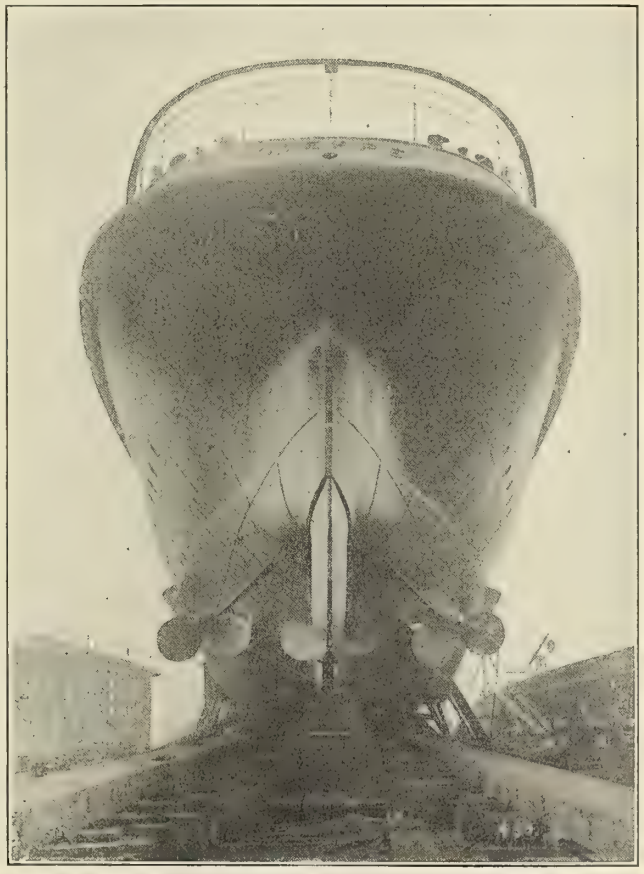
On the main deck forward there is the first-class dining saloon, which extends right across the ship, and has dining accommodation for 62 passengers. The officers' rooms, galleys, pantries, etc., are arranged on the main deck amidships. On the main deck aft is the second-class passenger accommodation. The sleeping saloons are on the lower deck, sleeping accommodation being provided for over 200 passengers.

The Turbines.—The propelling machinery consists of three independent Parsons' compound steam turbines and two condensers. There are two high-pressure turbines and one low-pressure turbine, and with the former are incorporated the reversing turbines. Each of the three turbines drives a separate shaft, with one three-bladed propeller on each shaft. The condensing water is circulated through the condensers by two large centrifugal pumps, one for each condenser, worked by independent engines. The centrifugal pumps are connected to large valves leading to the bilge, so that in the event of accident to the ship these pumps could be utilized for pumping out the engine room. The latter is fitted with all modern appliances, including a large feed filter and a suitable feed heater.

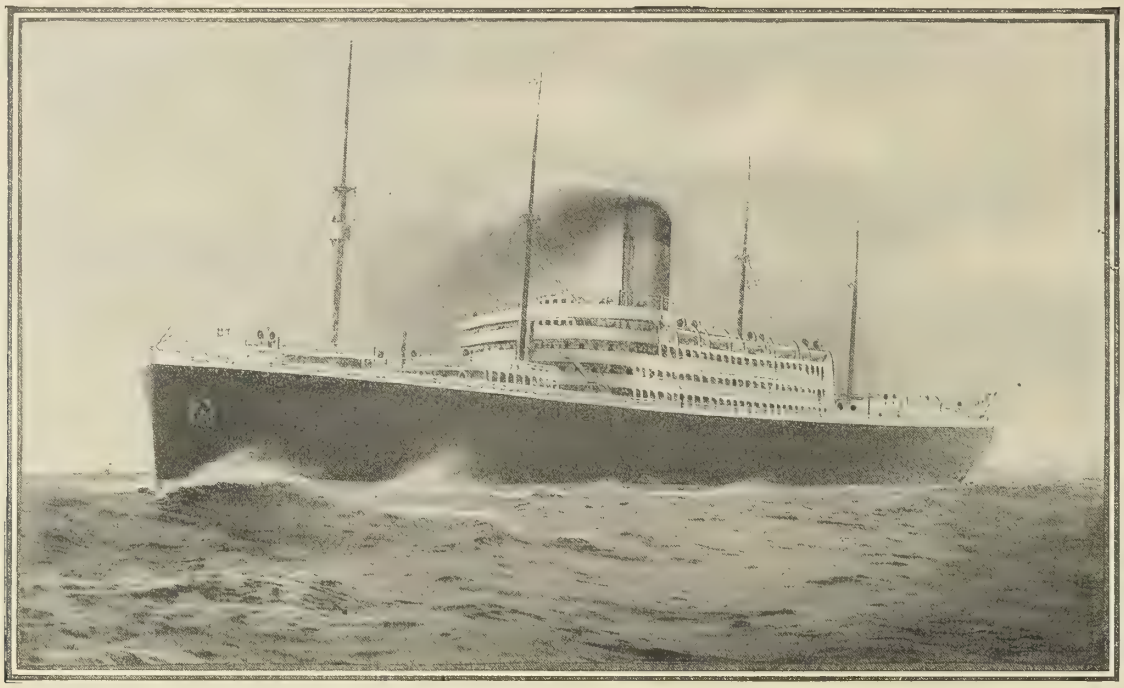
The boilers are four in number, and are single-ended, of the ordinary multitubular marine type, and fitted with Howden's forced draft. They are constructed entirely of steel and adapted for a working pressure of 150 pounds per square inch. The turbines and other machinery have been constructed by the Fairfield Company.

ALLAN MCPHERSON, JR.

THE DIEPPE READY TO BE LAUNCHED.



THE STERN, SHOWING THREE SCREW PROPELLERS.

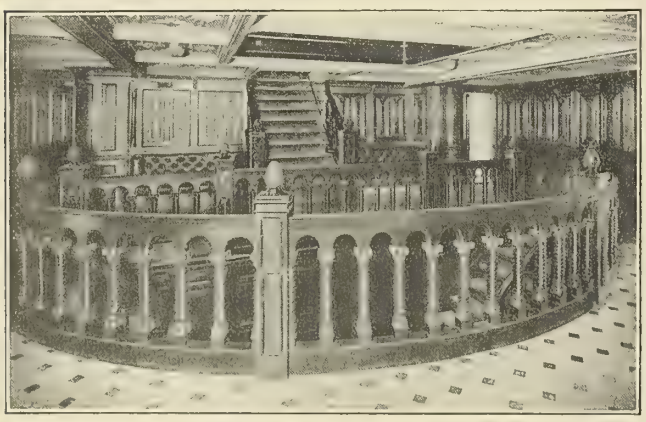


THE STEAMSHIP DAKOTA.

THE GREAT NORTHERN STEAMSHIP DAKOTA.

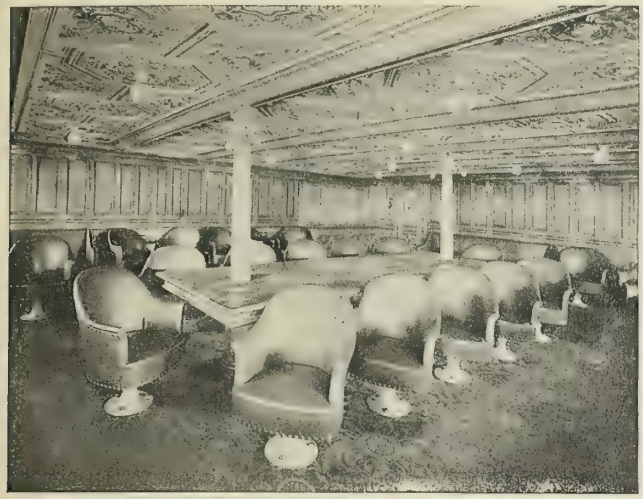
With the sailing of the *Dakota* from this port, near the end of April, the last large ship constructed for the American merchant marine for over-sea trade has passed out of the hands of the shipbuilders, and for the first time in a good many years there is not now under construction a single American ship for trans-oceanic commerce. The contract for the construction of the *Dakota* and her sister ship *Minnesota,* which is now in service on the Pacific, was let in June, 1901, and was the last contract ever awarded an American firm for ships for this trade.

The *Dakota,* which is among the largest ships afloat, measures 630 feet over all, with an extreme beam of 72 feet 6 inches, and a depth from the upper navigating bridge to the bottom of the keel of 88 feet 4 inches; the depth of the hull being 56 feet at the side of the ship, and 57 feet 6 inches at the center line. The load displacement at a draft of 34 feet is about 33,000 tons, while the gross tonnage of the ship is 20,718. This tonnage may be compared with the 21,000 of the *Caronia* and *Carmania,* the 23,500 of the *Baltic* and her new sister ship *Adriatic,* the 20,900 of the *Celtic* and *Cedric,* and the 22,500 of the new Hamburg-American steamers now completing at Belfast and Stettin respectively. The ten ships included in these figures are, we believe, the only ships afloat of above 20,000 tons. It is thus seen that the *Minnesota* and *Dakota* lack very little of being the largest ships in any sea, and it is claimed that when the cargo-carrying capacity is taken as a basis for measurement they are without any exception the largest vessels in existence. The illustrations which we present, showing both exterior and interior views, will give some idea of the mammoth size of these ships, the bow view in particular showing the immense freeboard, which is very striking as the ship is viewed from the pier. It must be remarked in passing, however, that at the time this view was taken the ship was not loaded to

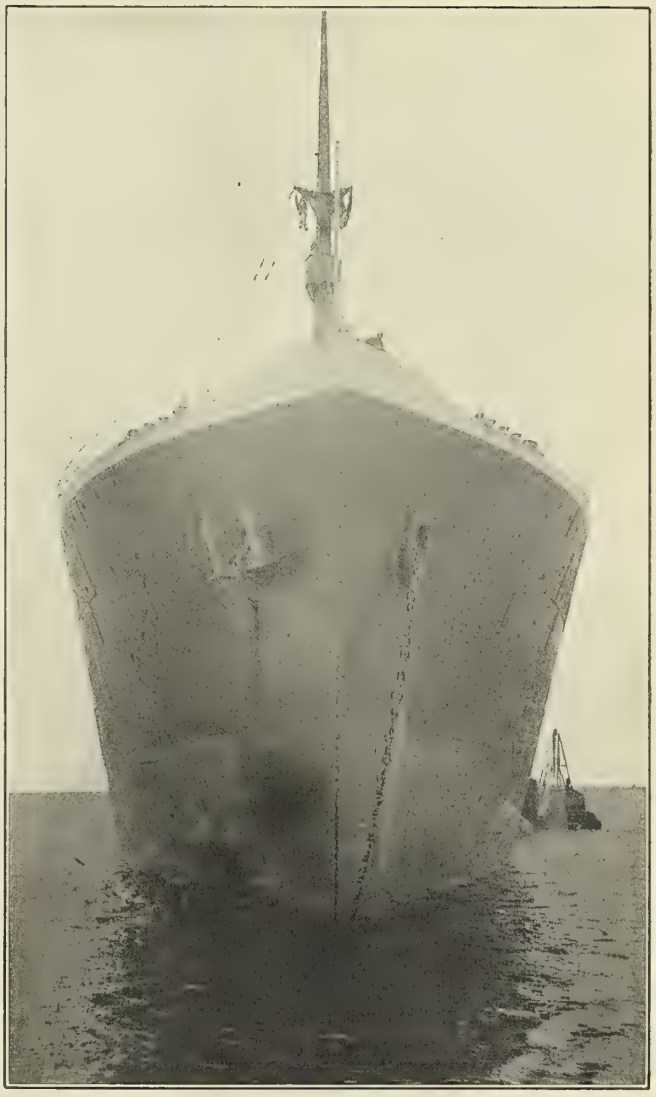


STAIRWAY IN SOCIAL HALL.

FIRST-CLASS SMOKING ROOM.



THE LIBRARY.



BOW VIEW OF THE DAKOTA.

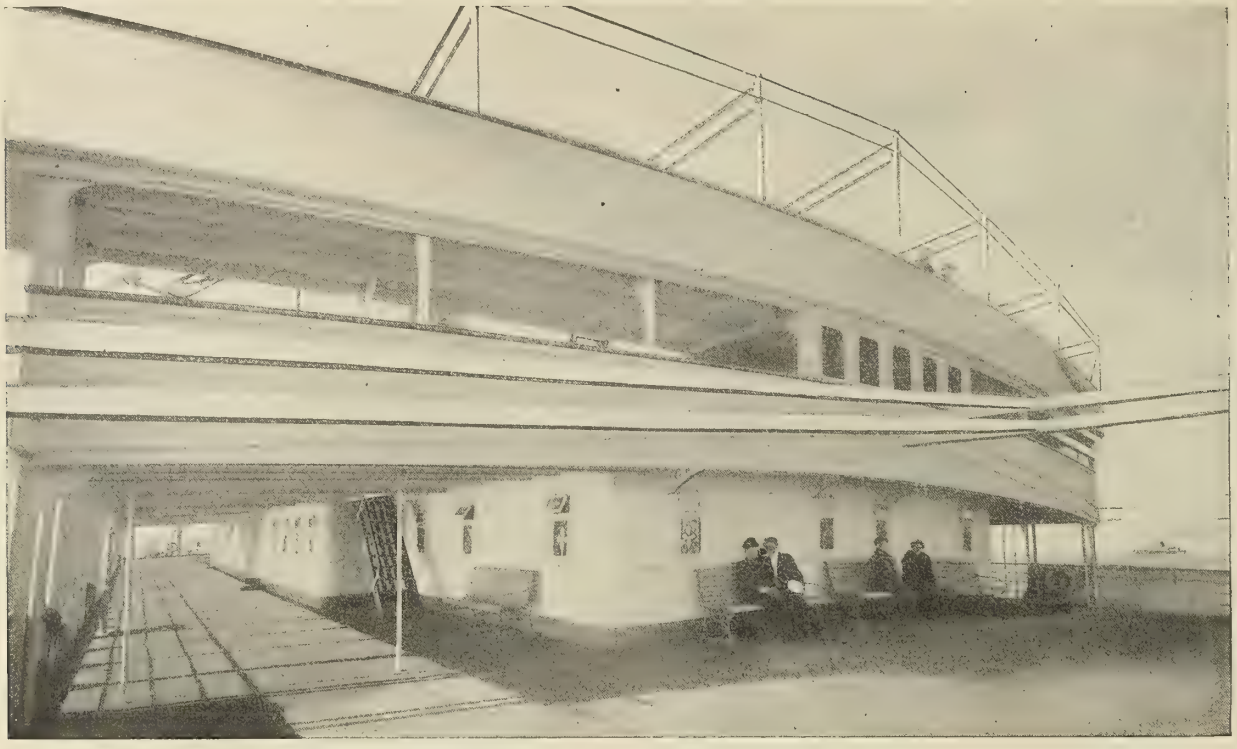
her full draft. Accommodation is provided for about 3,000 passengers, of whom 300 are first-cabin, about the same number second-cabin, and 2,400 third-cabin passengers. If it should be necessary to convert the ship into a troopship, the space devoted to third-class passengers is capable of accommodating 1,300 troops, or, on the other hand, it can be transformed into space for carrying cattle.

The first-cabin passengers are accommodated in the large deck house amidships, and our illustrations show that the accommodations provided for them are most comfortable and luxurious without being in any way gaudy or over-decorated. The large dining saloon, finished in mahogany, will seat about 200 passengers, and is situated at the extreme forward end of the immense deck house, having large light ports on three sides. Above the dining saloon the library is situated on one side of the ship and the ladies' boudoir on the other, both being very attractive quarters for the gathering of passengers in stormy weather. Nearly all of the first-class state rooms have outside windows, while many of them are arranged en suite with bath rooms attached. On the bridge deck is the smoking room, finished in red leather and Flemish oak, while below is a café in which meals may be obtained at all hours of the day and night.

For taking care of the commissary equipment and stores an immense refrigerating plant is installed, operated on the American-Linde system, which is capable of cooling the 300 tons of the ship's provisions in the shape of groceries and vegetables, meat, dairy products, etc., and also of refrigerating as much as 1,700 tons of cargo in the shape of pressed beef. Four large evaporators

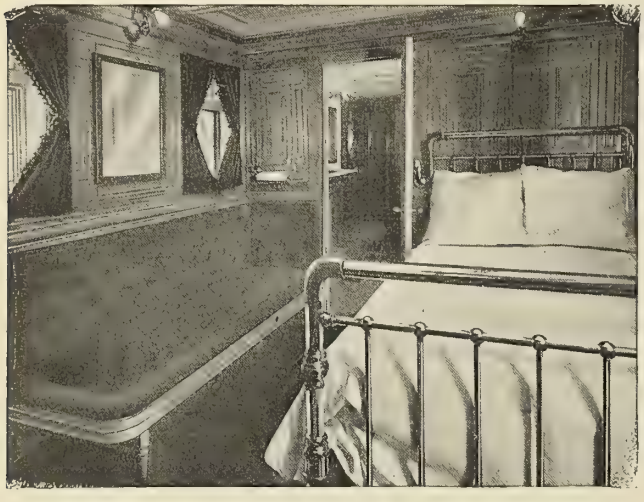
CAFÉ ON MAIN DECK.

A CORNER OF THE DINING SALOON.



THE PROMENADE DECK, SHOWING GREAT BREADTH OF SHIP.

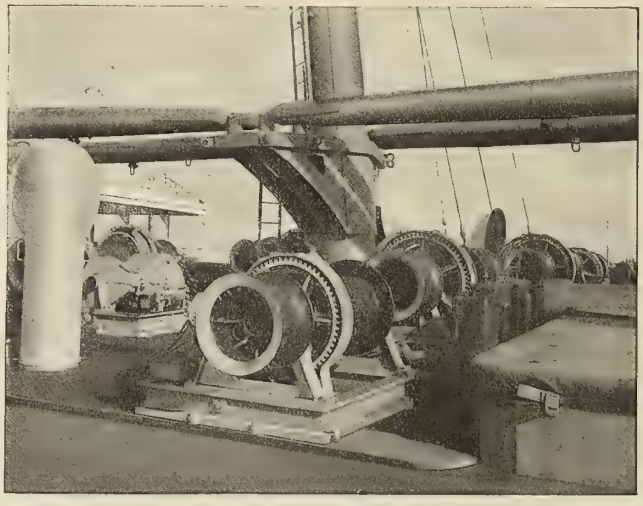
for furnishing fresh water are located in the engine rooms, and provide an abundance of water for both boilers and the use of the crew and passengers. These evaporators have a combined capacity of about 30,000 gallons of fresh water a day. The electric-light plant is very extensive, electricity being used not only for lighting all parts of the ship but for heating the state rooms, running the numerous ventilating fans, and supplying the power for both steering the vessel and operating the cargo-hoisting apparatus. This latter has been so designed as to do away with a large percentage of the men usually found on the deck of a ship when cargo is being hoisted in or out.



ONE OF THE SUITES DE LUXE.

The ship is driven by twin-screw, triple-expansion engines of about 10,000 combined horsepower, and operated by steam at a pressure of 250 pounds, generated by a battery of sixteen Niclausse water-tube boilers. Each engine is located in its separate water-tight compartment, while the boilers are in four similar compartments. One battery of four boilers has been provided experimentally with Duluth stokers manufactured by the Whiting Foundry and Equipment Company, of Harvey, Ill.

On trial trip, for a distance of 59 knots, the *Dakota* required 3 hours and 20 minutes, giving an average speed of 17.7 knots per hour. On this trial the revolutions of the engine were 87 per minute and the total indicated horsepower 10,874. The pressure of steam in the boilers was 245 pounds, and at the engine 215 pounds. Pocahontas coal was used, the consumption figuring out



ELECTRIC CARGO WINCHES.

at 1.75 pounds per horsepower hour. The speed of motion of the automatic grates of the Duluth stokers was 7 1-2 inches per minute. The draft of the ship forward was 17 feet 1 inch; aft, 21 feet 9 inches; mean, 19 feet 5 inches. The diameter of the two propellers is 20 feet, with a pitch of 22 feet.

The American Society of Refrigerating Engineers has issued its first year book, containing constitution and the various by-laws in force up to date. Copies of this book may be obtained by addressing the secretary at Suite 806, No. 258 Broadway, New York.

The Scranton meeting of the American Society of Mechanical Engineers will be held from June 6 to 9 inclusive. The professional papers which will be presented number seventeen.

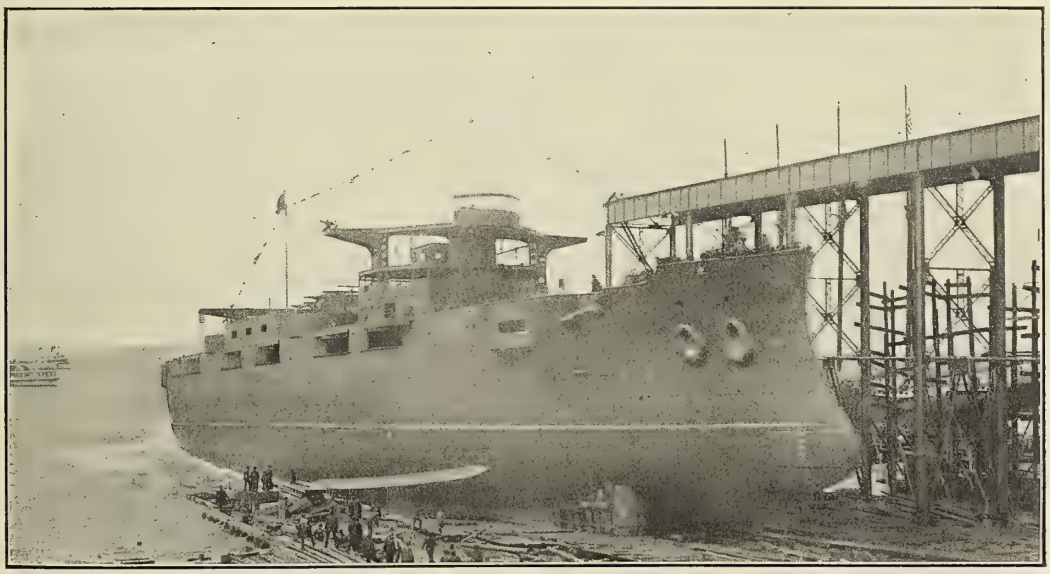
BREAKING THE BOTTLE ON THE BOW OF THE ST. LOUIS.

(Photograph by W. H. Rau.)

TWO NOTABLE LAUNCHES.

On Saturday, April 8, there was launched from the yards of the Newport News Shipbuilding and Dry Dock Company the new United States battleship *Minnesota,* of 16,000 tons, the ceremony being performed by Miss Rose Marie Schaller. This ship is one of several of similar type now under construction for the United States Navy, of which two, the *Louisiana* and *Connecticut,* have been described in Marine Engineering, at pages 450, of October, 1904, and 489, of November, 1904, while their engines were described at page 198 of April, 1903. As we have already described the ship at some length, we will give here simply a *resumé* of the principal dimensions and features.

The *Minnesota* has a length of 450 feet, beam of 76 feet 10 inches, and at a draft of 24 feet 6 inches will displace about 16,000 tons. Equipped with twin-screw machinery of an aggregate of 16,500 horsepower the speed on trial is designed to be 18 knots. Her normal coal supply is 900 tons, which can be increased when required to a total of 2,200 tons. Steam is furnished by a battery of Babcock & Wilcox boilers. The steaming radius at 10 knots is expected to be 5,000 nautical miles. The battery consists of four 12-inch and eight 8-inch rifles mounted in pairs in six turrets, with twelve 7-inch rifles firing through ports on the gun deck; and a secondary battery of twenty 3-inch rapid-fire guns, twelve 3-pounders, eight 1-pounders, eight machine guns, and two field guns. There are also fitted four submerged torpedo tubes of the new 21-inch type. The water-line belt of armor is of a uniform



THE MINNESOTA ENTERING THE WATER.

thickness amidships of 9 inches, tapering to 4 inches at the bow and stern. The barbettes for the 12-inch guns have thicknesses of 10 and 7 1-2 inches in front and back respectively, while those for the 8-inch guns have thicknesses of 6 and 4 inches. The turrets for the heaviest guns have 12-inch port plates with 8-inch sides and backs, while the corresponding thicknesses for the 8-inch turrets are 6 1-2 and 6 inches. The protective deck has a thickness of 3 inches on the slopes and 1 1-2 inches on the flat. The ship will be manned by a crew of 41 officers and 815 men. The contract for her construction was signed June 20, 1903, the price being $4,110,000; the contract date of completion is December 20, 1906.

The other launch which we illustrate is that of the cruiser *St. Louis,* which left the ways of the Neafie & Levy Company, at Philadelphia, on May 6, the sponsor having been Miss Gladys Bryan Smith, of St. Louis. This ship is a semi-armored cruiser,

THE LAUNCH OF THE ST. LOUIS.
(Photograph by W. H. Rau.)

and is identical in design with the *Milwaukee* and *Charleston,* now being completed at the Union Iron Works of San Francisco and the Newport News Shipbuilding and Dry Dock Company respectively, and described at page 474 of Marine Engineering for October, 1904.

The length is 424 feet, beam 66 feet, and displacement on a draft of 22 feet 6 inches is 9,700 tons. She is propelled by twin-screws actuated by vertical triple-expansion engines of the four-cylinder type, with cylinders respectively 36, 59 1-2, 69, and 69 inches in diameter and a stroke of 54 inches. At 133 revolutions per minute these engines are designed to develop 21,000 horsepower and drive the ship at a speed of 22 knots. Steam is furnished by a battery of sixteen Babcock & Wilcox boilers. The normal coal supply is 650 tons, which may be increased when necessary to a total of 1,500 tons. The main battery consists of fourteen 6-inch rapid-fire guns distributed upon the upper and main decks, while the secondary battery contains eighteen 3-inch rapid-fire guns, twelve 3-pounders, twelve 1-pounders, ten machine guns, and two field guns. A protective deck is fitted, having a thickness on the slopes of 3 inches and on the flat of 2 inches, while 4-inch vertical side armor is fitted for a considerable length along the midship portion of the vessel. The complement of officers and men numbers 670 all told. The contract for the construction was signed March 11, 1901, the price being $2,740,000.

Wisdom of the Amalgamation of the Line and Engineer Corps of the Navy.

Editor Marine Engineering:

There has been a great deal said against the amalgamation of the line and engineer corps of the navy, which took place by act of Congress in 1899, but if the subject is carefully studied it will be seen that it is the best arrangement.

The engineer corps was not abolished and the standing of the engineer was not degraded, as has been said, for the line officer is now an engineer officer. Those who are not competent can qualify themselves in either branch, if below forty years of age; and all the younger ones can easily be given a training which will make them competent. Every commissioned officer doing deck or engine duty on a ship should be trained to do either in case part of the complement is killed in action.

No commanding officer who is not an engineer has the knowledge which will enable him to get the best results out of his ship. Too often do commanding officers, who are without engineering training, treat the engine department as if it did not belong to their ship at all, and instead of helping along the officer in charge of it, many obstacles are often thrown in his way, and it is very hard to keep the department in a thoroughly efficient condition. All commanding officers should know what can be gotten out of the machinery and the men of the engineer force under different conditions, and also they should thoroughly understand the difficulties to be surmounted in overhauling and making repairs and keeping the department in a thoroughly efficient condition. They would then certainly consider the engine department as much a part of their ship as any other, and would always lend it a helping hand.

There was no mistake made in the amalgamation of the line and the engineer corps of the navy. What the navy needs is more officers. Those that desire to do so can qualify for engineering duty only. When there are sufficient commissioned officers to give each ship the number for engineering duty that she needs there will be a great improvement. A battleship should have a chief-engineer and six assistants, commissioned officers, instead of a chief-engineer and one assistant, as is the case at present. This shortage of commissioned officers has thrown great strain on those doing engineering duty on the ships.

The warrant machinists are a worthy class; there are doubtless some exceptions, but the greater part of them have neither the education nor the training to fit them for engineer officers. An educated engineer officer is necessary to obtain the best results.

There is really nothing necessary to the proper running of a ship in either deck or engine department that cannot be accomplished with application, by an educated person brought up to sea life. If one has never seen a ship or a machine shop before, the United States laws allow him to (1) qualify as master of ocean steamers in five years, or (2) qualify as chief-engineer of ocean steamers in six years, in the merchant service. Those who wish to specialize along the lines of designing and building machinery and ordnance, and otherwise, can do so, as particular aptitude shows itself. As years go by the mechanical appliances on board ship are increasing. Every part of a gun is a machine. Everything is done by machinery. Seamanship has gone, and all that remains is handling a ship under steam, and the navigation. What is needed now is engineers with training in navigation and in handling steam vessels. The all round training and manual dexterity of Admiral Cochrane, R. N., who could show every man aboard, except the doctor, how to do his work, is as much needed now as it was then.

A Chief-Engineer not in the Navy.

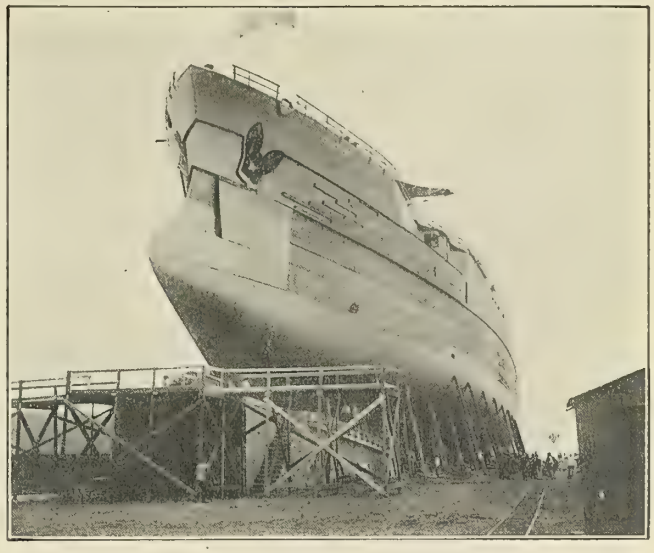
The Launch of the French Battleship Liberté.

On the 19th of April the latest of the French battleships was launched from the yards of the Ateliers et Chantiers de la Loire, near St. Nazaire. This ship is one of the *République* class of six ships, which will form a very powerful squadron in the French fleet, and will be distinctive from most of the other ships in that fleet, not only on account of the fact that they are the largest ships yet designed for the French navy, but because of the fact that they are identical in design, whereas preceding ships have in almost every instance been of heterogeneous design, even though built at the same time and given approximately the same elements of military value. Being one of the latest of the *République* class, however, there are a number of improvements incorporated in the *Liberté* which are more in the nature of details than in the main features of the design. One alteration, however, which is of great importance, is the substitution of ten 7.7-inch guns for the sixteen 6.4-inch guns in the earlier ships. The *Liberté* embodies a thoroughly well-studied compromise of all the requirements imposed upon the designer of a modern battleship.

In order to institute a comparison between the principal data of this ship and those of the latest battleships of other powers, there is appended a table giving the principal characteristics of one battleship from each of the seven leading naval powers of the world, from which it is seen that the *Liberté* makes a very com-

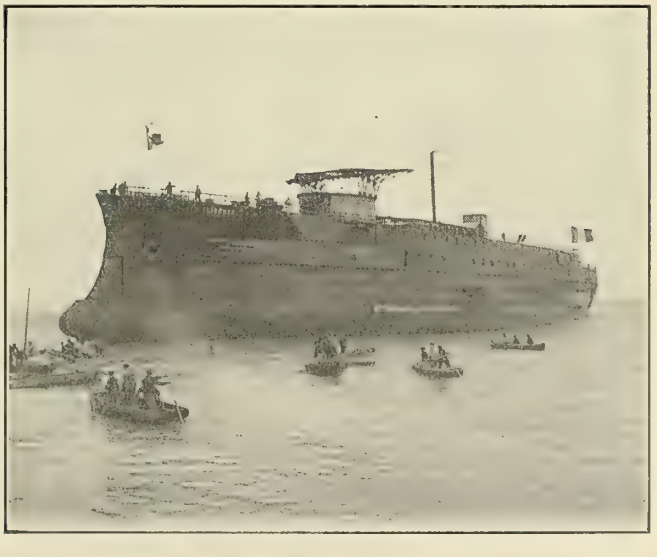
Navy	England.	United States.	France.	Germany.	Italy.	Russia.	Japan.
Ship	King Edward VII.	Kansas.	Liberté.	Deutschland.	Vittorio Emanuele III.	Imperator Paul I.	Kashima.
Displacement	16,350	16,000	14,750	13,000	12,630	16 630	16 400
Length, L. W. L.	425′	450′	440′	400′	435′	425′	455′
Beam, extreme	78′	76′ 10″	79′6″	72′	73′6″	78′	78′2″
Draught, mean	26′9″	24′6″	27′6″	25′9″	25′10″	27′	26′4″
Horsepower	18,000	16,500	*18,000	*16,000	*21,000	18,000	21,500
Speed in knots	18.5	18	18	18	21	18	18.5
Armor, water line	9	9	11	9	10	11	9
" for'd, aft	6-3	4-4	6-6	7 and 4	6 and 4	6-6	6.5-6.5
" citadel	8	7	10	5.5	8	?	6
" casemate	7	7	6	6.75	None	None	6
" deck	2	3	2-3	3	4	2.75-4	3
" barbettes	12	10	11	10	10	12	9
" turrets	12-7	10-6	12-5	11	10-6	12-7	9-5
Main battery	4-12″ 4-9.2″ 10-6″	4-12″ 8-8″ 12-7″	4-12″ 10-7.7″	4-11″ 14-6.7″	2-12″ 12-8″	4-12″ 12-8″	4-12″ 4 10″ 12-6″
Secondary battery	14-12 lbr. 14-3 lbr. 2 mach.	20-3″ 12-3 lbr. 8-1 lbr. 12 mach.	26-3 lbr. 16-1 lbr.	22-3.4″ 6-3 lbr. 16 mach.	12-3″ 12-3 lbr. 4 mach.	?	12-12 lbr. 12-3 lbr. 6 mach. .....
Torpedo tubes	5-18″	4-21″	5-18″	6-18″	4	4	5
Coal supply	950-2,000	900-2,200	900-1,850	800-1,800	1,000-2,600	?	1,200-2,000
Crew	800	800	750	700	700	900	980

*3 screws.



THE BOW, SHOWING ENORMOUS RAM.

THE STERN, SHOWING THREE-SCREW SHAFTS.



THE LIBERTÉ AFLOAT.

mendable showing in comparison with most of the others. If we compare her with the *Deutschland* of the German navy, or the *Kansas* of the United States navy, it is seen that her powerful guns could inflict serious damage upon the relatively thin water-line belts of these hypothetical enemies, while her own 11-inch armor belt would render her almost invulnerable to shells from the other ships. The same holds true when we compare her with the English *King Edward VII* and the Japanese *Kashima,* except for the fact that the two latter ships have one knot greater speed, and could therefore engage or not at will, or could choose the distance best suited to themselves in case an engagement was found desirable.

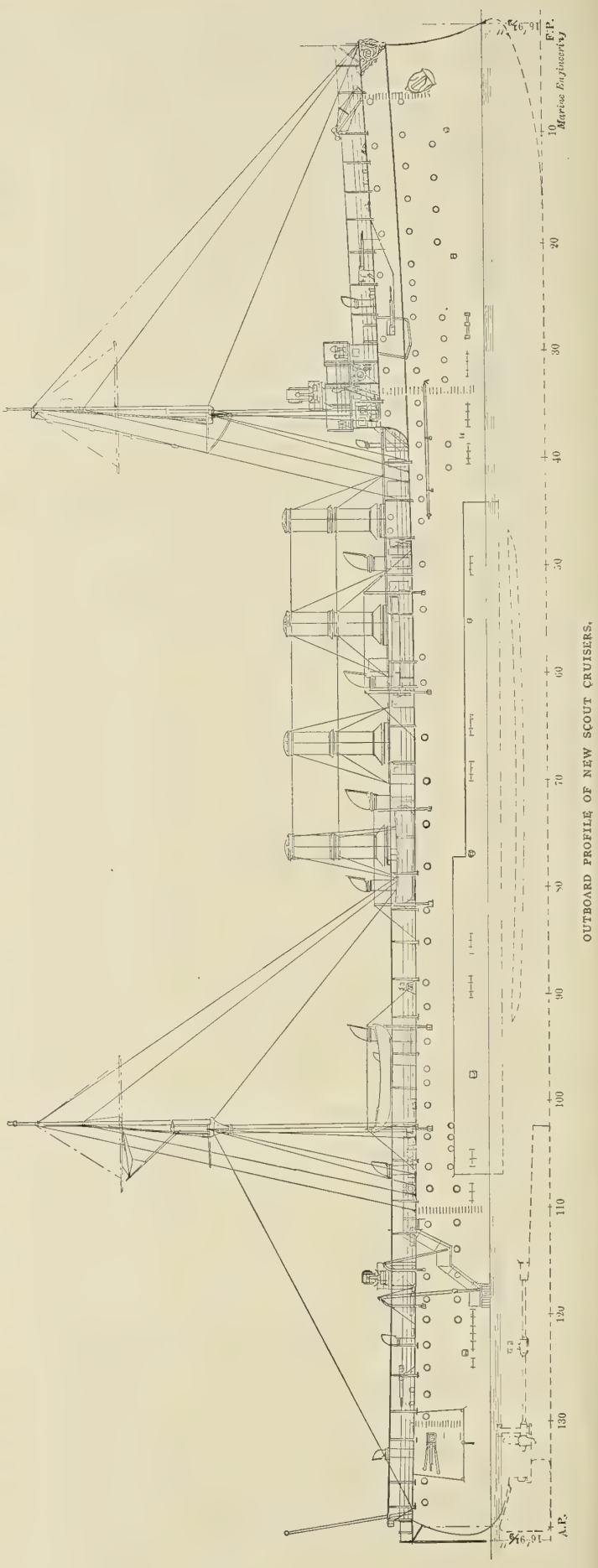
-J. PELTIER.

THE NEW UNITED STATES SCOUT CRUISERS.

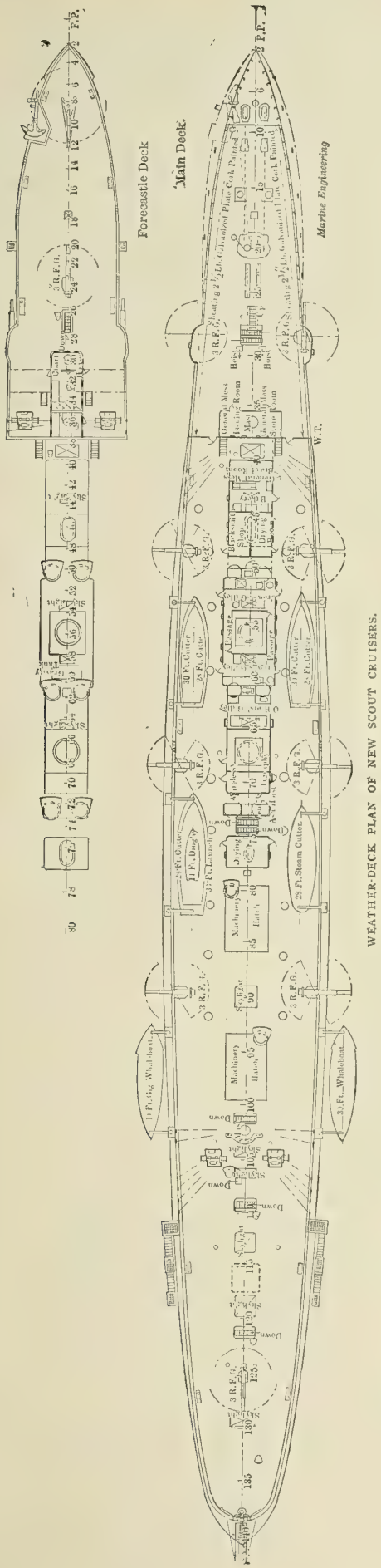
Bids for the construction of three new cruisers for the United States navy, to be known respectively as *Birmingham, Chester,* and *Salem,* were opened by the Secretary of the Navy April 15, and a very large number of bids was found to have been submitted. Five of the large shipbuilding companies on the Atlantic coast, together with the Union Iron Works, of San Francisco, presented bids for constructing the cruisers on the department's plans, with reciprocating engines and twelve water-tube boilers; and the five Atlantic yards submitted alternate bids for the construction of the ships on their own plans, fitted with steam turbines and with varying equipments of water-tube boilers. The successful bidders turned out to be the Bath Iron Works for one ship propelled by Parsons steam turbines, to be constructed in thirty-six months, with a guaranteed speed of 24 1-2 knots, at a price of $1,688,000; and the Fore River Shipbuilding Company, of Quincy, Mass., for two ships, one of which is to be propelled by reciprocating engines and to be completed in thirty months, while the other is to be propelled by Curtis steam turbines and Normand water-tube boilers, each to be constructed at a price of $1,556,000. These were not the lowest bids submitted in the several departments, but the combined features of price and design were considered by the department to embody the best combination of features submitted.

Through the courtesy of the Navy Department we are enabled to present an outboard profile and deck plan of these cruisers, giving a very good idea as to the general appearance which they will have when completed, and the arrangement of the battery of 3-inch guns. The ships have a length between perpendiculars of 420 feet; a beam, molded, of 46 feet 8 inches; and on a draft of 16 feet 9 1-2 inches will displace 3,750 tons. On this displacement, which is that at which the speed trial will be run, they will carry 475 tons of coal and 50 tons of feed water, the total bunker capacity for coal being 1,250 tons. Each ship is provided with a battery of twelve 3-inch rapid-fire guns, mounted as shown in the illustrations, together with two submerged torpedo tubes of the new 21-inch type, which are the most powerful weapons of this sort yet designed.

The engines under the department's plans will be of the twin-screw, four-cylinder, triple-expansion type, with a combined indicated horsepower of 16,000, with the propellers arranged for outboard turning when going ahead. The steam pressure will be 250 pounds per square inch, the stroke will be 36 inches, cylinder diameters respectively 28 1-4, 45, and (two) 62 inches, and the revolutions per minute at full power about 200. Each engine will, as usual, be located in a separate water-tight compartment. The twelve water-tube boilers will be of the express type, placed in three water-tight compartments, and having a total grate surface of 693 square feet and a heating surface of 37,080 square feet, this giving a ratio of heating surface to grate surface of 53.4 to 1. The working pressure in the boilers will be 275 pounds per square inch. The four funnels will have a height of about 75 feet, and the draft provided by them will be supplemented when necessary by fan pressure amounting to about 5 inches of water. The steaming radius at 10 knots per hour is expected to be about 6,250 knots, and at full speed about 1,875 knots.



OUTBOARD PROFILE OF NEW SCOUT CRUISERS.



WEATHER-DECK PLAN OF NEW SCOUT CRUISERS.

In order to insure the ability of these ships to make high speed at sea, and to maintain their speed during all conditions of weather, they have been given very considerable freeboard, amounting at the bow to not less than 34 feet, at the stern to 21 feet 6 inches, and amidships 19 feet 8 1-2 inches, these figures being based on normal draft. This freeboard is considerably higher than that which has been given to any other ship in the United States navy, being even greater than that of the *Brooklyn*, which was at the time of her design higher out of the water than any other warship in existence. This fact will render the scout cruisers quite easy of detection by an enemy at sea, and is in striking contrast to the practice inaugurated by the British navy in the design of the eight scout cruisers now approaching completion for that service. The latter ships resemble very closely an enormous torpedo-boat destroyer, being relatively low in the water and comparatively free from the superstructure features so commonly in evidence on ships of a large size. It remains to be seen whether or not this quality of invisibility possessed by the British cruisers will have a greater military value than the extreme sea-keeping qualities of the American cruisers, together with the fact that their lookouts, being so high above the level of the sea, will have a much greater range of vision than with the lower ships.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, report under date of May 10, 1905, the following percentage of completion of vessels building for the United States Navy:

			Apr. 1.	May 1.
BATTLESHIPS.				
Virginia	19 knots.	Newport News Co.	83.74	85.53
Nebraska	19 "	Moran Brothers Co.	69.31	70.89
Georgia	19 "	Bath Iron Works.	76.44	77.78
New Jersey	19 "	Fore River Shipbuilding Co.	79.4	81.1
Rhode Island	19 "	Fore River Shipbuilding Co.	84.	86.
Connecticut	18 "	Navy Yard, New York, N. Y.	70.24	73.56
Louisiana	18 "	Newport News Co.	71.9	74.83
Vermont	18 "	Fore River Shipbuilding Co.	43.4	47.2
Kansas	18 "	New York Shipbuilding Co.	48.1	51.4
Minnesota	18 "	Newport News Co.	61.76	63.18
Mississippi	17 "	Wm. Cramp and Sons.	24.39	27.04
Idaho	17 "	Wm. Cramp and Sons.	22.27	24.93
New Hampshire	18 "	New York Shipbuilding Co.	1.5	3.2
ARMORED CRUISERS.				
California	22 knots.	Union Iron Works	73.5	75.
Maryland	22 "	Newport News Co.	99.	99.95
South Dakota	22 "	Union Iron Works.	71.3	72.9
Tennessee	22 "	William Cramp and Sons.	69.45	71.8
Washington	22 "	New York Shipbuilding Co.	70.2	74.2
North Carolina	22 "	Newport News Co	2.15	3.25
Montana	22 "	Newport News Co	2.01	3.2
SEMI-ARMORED CRUISERS.				
St. Louis	22 knots.	Neafie and Levy Co.	60.6	63.
Milwaukee	22 "	Union Iron Works.	69.8	72.4
Charleston	22 "	Newport News Co.	92.56	93.78
GUNBOATS.				
Dubuque	12 knots.	Gas Engine and Power Co.	90.9	94.28
Paducah	12 "	Gas Engine and Power Co.	82.8	83.9
TRAINING SHIPS.				
Cumberland	Sails.	Navy Yard, Boston.	95.	95.
Intrepid	"	Navy Yard, Mare Island	95.	97.5
TORPEDO BOATS.				
Goldsborough	30 knots.	Wolff and Zwicker.	99.	99.
O'Brien	26 "	Lewis Nixon.	99.	99.

Albert S. Crane has become chief hydraulic engineer of the J. G. White Company, of New York, and will be in responsible charge of the hydraulic department.

Messrs. Sadler, Perkins & Field, marine engineers and contractors, have been incorporated under the same style.

James D. Lamb, of the Lamb Boat and Engine Company, of Clinton, Iowa, was drowned on May 12 in the river near Clinton.

Captain Worth G. Ross, of the revenue cutter service, has just succeeded as chief of that service Captain Shoemaker, retired.

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Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

The edition of the June issue of Marine Engineering comprises 5,500 *copies. We have no free list, accept no return copies and issue only enough to supply the regular demand, so that the* 5,500 *copies represent legitimate circulation.*

Power Boats.

We take pleasure in announcing that arrangements have been concluded with Professor W. F. Durand for a series of articles upon the general subject of power boats, of which the first will be published in the July number of Marine Engineering. These articles are to include the entire problem of design and operation, from the laying of the keel to the completion of the craft; and will also look after the management of both boat and machinery when under way. There will also be included observations upon handicap time allowances for racing, as dependent upon both hull and machinery. The series should prove very valuable to designers and owners alike.

Engineers in the Navy.

In another column we publish a communication on the subject of the amalgamation of the line and engineer corps in the United States Navy, which is interesting as well for the large amount of truth it contains as for the considerable modicum of error. The writer of the communication is quite correct when he says that commanding officers in the navy who lack an engineering training are very likely to neglect the engineering departments of their ships, and to hamper with gratuitous obstacles the proper fulfilment of their duties by those charged with the operation of the vast amount of steam and other machinery fitted on board the modern warship. He is quite correct when he says that these commanding officers should have some measure of appreciation of the difficulties to be surmounted, in overhauling and making repairs to this machinery, and keeping it generally in a thoroughly efficient condition. He is very decidedly correct when he states that what the navy needs is more officers, and that the present shortage has thrown a great amount of strain on those now doing engineering duty on the ships.

He is decidedly in error when he states "there is really nothing necessary to the proper running of a ship in engine department that cannot be accomplished with application by an educated person brought up to sea life." The fact of this error is made manifest by the remark a little later down in the same paragraph to the effect that mechanical appliances on board ship are increasing, and that everything is now done by machinery. The Editor has seen cases where a man has gone through college—an educated person, in other words—and has endeavored to assimilate an engineering instinct by taking work in engineering departments under signally competent instruction, but has utterly failed to imbibe anything of that spirit, or to become in any sense a competent engineer. It may not be a thoroughly recognized fact, but it nevertheless *is* a fact, that engineers, like poets, are "born and not made." This fact in itself would militate very strongly against the whole theory promulgated by our correspondent in favor of letting officers perform indiscriminately line and engineering work, without any reference to the especial fitness of the officer in question for the particular work which he is called upon to perform. Another error of this same general character into which our correspondent falls, is when he states that "the all round training and manual dexterity of Admiral Cochrane is as much needed now as it was" in his time. It is entirely superfluous to say that the present is an age of intense specialization, and is daily becoming more so. It is entirely unnecessary to state that in a great profession like engineering the "Jack of all trades" has no place. When it comes to the proposition of handling the exceedingly intricate and delicate machinery outfit in the extremely restricted space allotted to it on a modern warship, most intense application to the particular requirements of the situation are absolutely necessary before any man, naval or civilian, is competent to handle the machinery, and get any sort of results out of it.

Our correspondent has little respect for the present occupants of the much overworked and much berated class known as "warrant machinists," a name which, by the way, is an entire misnomer, as the men bearing it

are, in fact, nothing more nor less than assistant engineers, and should be so denominated. If it is a fact, as he states, that "the greater part of them have neither the education nor the training to fit them for engineer officers," the fault lies, not with themselves, but with the system by which they are hounded and kept entirely subservient to men of the "cub ensign" type, who are in many cases of considerably inferior capacity and personality generally, but who may happen to have achieved a somewhat higher rank in the service—with the system which offers them no hope for the future, of attaining anything more desirable than the posts they now occupy. The great shortage of engineering officers has come about largely as a result of the navy personnel bill, which gives the former officers of this class the somewhat higher sounding appellations of "commander," "captain," etc., and places them in line duty rather than in the positions for which they were trained, thereby withdrawing from the engineering staff the very officers who are needed to keep up its status, and maintain conditions in such a state of efficiency as to make the ships of the navy individually and collectively fit for any service to which they may be called, at any time of the day or night, and under any circumstances which may arise.

If sufficient inducement can be offered to enable men from civil life, notably graduates of our engineering colleges, to enter the grade which is now known as that of "warrant machinist," but which, as above stated, ought to be known as "assistant engineer," with some chance of attaining ultimately the post of chief engineer, the problem might be solved in a satisfactory manner; but with the present conditions—with the inherent difficulty of the situation at the best, with the fact that even under the best possible circumstances the work is one requiring the worker to be in the engine room in the midst of dirt and oil for the greater part of his working day—it is little wonder that the number of applications from men educated as graduates of our engineering colleges are educated is very small, and is not likely to increase. Until some such idea is developed, and the berth made worth while for men of this character, it is idle to expect such men to enter it, or to expect any amelioration of the conditions which now exist. The question is one of extreme importance, involving as it does the very foundation of naval efficiency, and is one to which earnest attention should be given by those charged with the work of naval administration.

Marine Machinery.

As announced editorially two months ago, we present with this issue another installment of the articles on the "Design of Marine Machinery," by Professor Durand, and desire to announce that we hope to continue these installments from time to time, until the entire field is thoroughly covered.

The Ocean Yacht Race.

Delayed by heavy fog, the yachts entered in the transatlantic race were prevented from sailing on the date set, May 16, but succeeded in getting away the following day. Everything looks forward to a splendid trial of speed, as the entire eleven yachts crossed the line in less than an hour after the signal was given, and have kept pretty well together up to the date of writing. Speculation as to the probable winner would be fruitless, as the race will probably be decided definitely about the time this issue reaches the readers. With good winds and not too heavy a sea it is quite reasonable to expect that several of the yachts will reach their destination before the close of the month of May.

The Young Merchant and His Servant.

Once upon a time a young merchant just starting in his business career had a very efficient and thoroughly reliable servant upon whom he depended for a large amount of the business which came to him, which business was transacted in a splendidly efficient manner. The business prospered and both master and servant grew in experience and in general breadth of outlook, while the operations extended wonderfully in all directions. After some years had passed, however, the merchant having been gradually drawn away into various other fields of activity, came to neglect his useful servant in such a manner that the relations between the two were by no means the same as had been the case during the first years of their co-operation. Gradually and by almost insensible degrees the merchant began to make use of other servants, not his own, to the exclusion of the services of the one who had been so faithful; and it ultimately came about that he paid large sums of money to these foreign servants because he found that they would do the work for him a little bit cheaper than the one who had been accustomed to doing it, and gradually a large amount of his business passed under the control of these foreign agents. Meanwhile, the honest and faithful servant, not understanding the reasons for the condition of affairs which had become so detrimental to his own prosperity and to the independence of both himself and his master, had begun to languish; his efforts in behalf of his erstwhile indulgent master were no longer so whole-hearted as they had been in the happier days of the early part of the co-partnership. The business fell more and more into the hands of the successful foreign rivals, and things generally were in a very unsatisfactory state.

Thus affairs have continued to this very day, and the people of the United States continue to pay out vast sums of money, amounting to hundreds of millions of dollars, to aid in the upbuilding of the merchant marines of foreign countries, to the exclusion of that of our own.

THE DESIGN OF MARINE MACHINERY.*

BY W. F. DURAND.

PART V.

The Stephenson Link.

(2) *Block Slip.* Introductory to the discussion of the remaining items of a Stephenson link valve gear we must consider the subject of block slip.

Suppose, in Fig. 29, the link block to be made fast in the link at the end *A*, thus guiding or forcing this particular point of the link to move in the straight line *OAV*. Then the system will be completely constrained, and each point of the link will move in a definitely determined path, of which examples are given in Fig. 30, corresponding to various points along the link from one end to the other. Next, suppose that the block is cast loose, as in the normal working of the gear, and the point *R* is taken hold of and in some manner guided in the curved path corresponding to *R*, and as shown at *h*, Fig. 30. Then it will be clear that the point *A* on the link will move, as before, on the center line, and that consequently the link and block will move back and forth without relative motion between the two, or, in other words, without slip of the block.

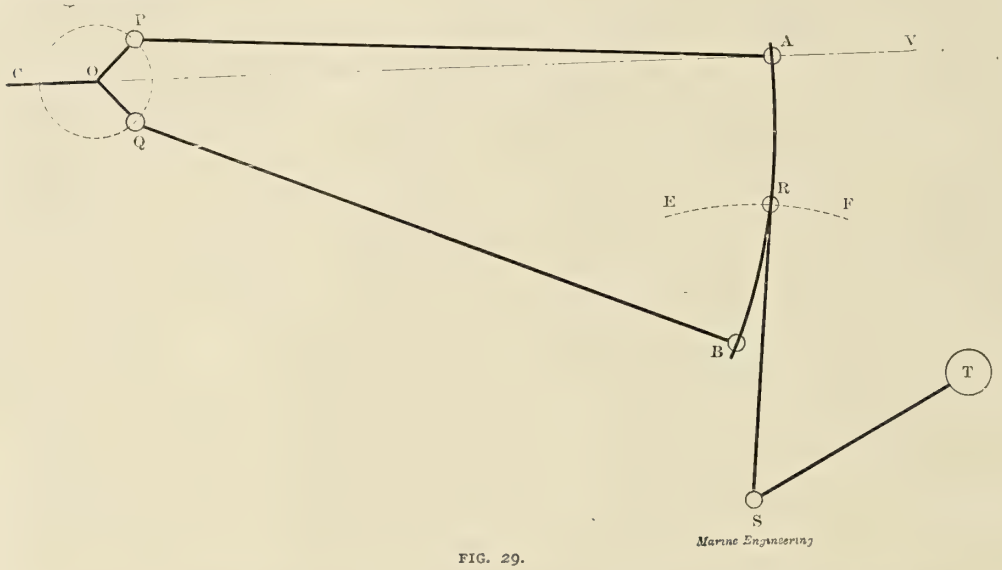


FIG. 29.

Suppose, on the other hand, that the link is controlled by a bridle rod *RS*, and that by reason of the location of the center *S*, this point is forced to move in the arc of a circle, as indicated by *EF*, Fig. 30. Then it will be clear that the point *A*, Fig. 29, cannot now move in a straight line, but must move back and forth across the center line, departing from a straight line path to an extent equal to that by which the curve *h*, Fig. 30, departs from the arc *EF*. Since the link block, however, must move on the center line, it will follow that the operation of the gear will be accompanied by a sliding of the link back and forth on the block, thus producing a relative sliding of the block to and fro in the link.

Inasmuch as the usual means of controlling the link is by a bridle rod such as *RS*, it will be clear that its operation will thus be accompanied by such relative motion of the block and link, depending in amount on the details of the design and the proportions of the gear.

It must not be assumed that block slip is in itself necessarily undesirable, and that it must at all hazards be reduced to the lowest possible limits. If excessive it may result in undue wear on both link and block, and the vibration of the link to and fro may enter as a factor into the problem of ship vibration. Block slip, however, should be considered simply as one of the factors in the operation of the gear, and its influence for good and ill must be weighed with that of other items, and holding in view the general purposes to be fulfilled by the problem in hand. In particular, as we shall proceed to show, the fact of block slip may, if desired, be made use of to produce certain modifications in the movement of the valve having relation to the balancing up of the events in the two ends of the cylinder.

By reference to Fig. 16† it will be remembered that in the operation of a gear with single

*Continued from February, 1904, page 94.

†See MARINE ENGINEERING, November, 1903, page 591.

eccentric there is a tendency on the up stroke toward relatively earlier cut-off and earlier events generally, and on the down stroke toward relatively later cut-off and later events generally. Also, by reference to the tabular statement in [11] it will be remembered that linking up the gear gives earlier events, and with the open-rod configuration an increasing lead, while linking out the gear gives later events, and with open rods a decreasing lead.

Suppose, then, that the block slip were such that when the piston is near the point of cut-off on the down stroke the link is pushed over so as to bring the block nearer the middle relative to its mean or average position, while on the up stroke it is forced over in the other direction so as to carry the block nearer the end relative to the mean position. Then it will be clear that in such case there will be a tendency toward a relatively earlier cut-off on the down stroke and later cut-off on the up stroke than would result if the block were fixed in the link at its mean position for the revolution. Such influence would therefore tend to counteract the later cut-off on the down stroke and earlier cut-off on the up stroke, which would result with the block thus fixed. If sufficient in amount this modification may thus result in a very satisfactory equalization of the points of cut-off for the two strokes.

Again, it is usually considered desirable for the better control of the moving parts to give to the valve an earlier opening and greater lead at the bottom of the stroke than at the top. It is seen that this condition may be realized as a result of block slip provided the arrangement is such as to force the link over into a relatively linked up position when the piston is at the bottom of the stroke, and into a linked out position when at the top, estimated each with reference to the mean position for the revolution.

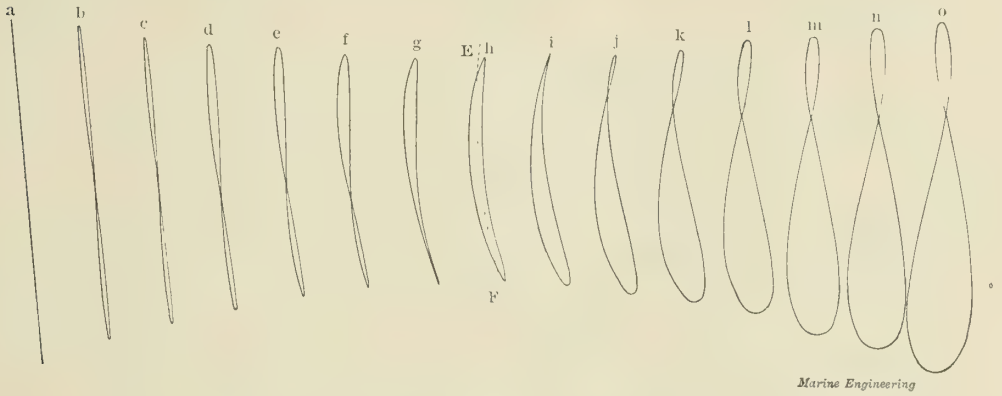


FIG. 30.

It is, of course, true that such changes in point of cut-off and in lead may be brought about by making the steam lap quite unequal on the two ends of the valve, giving less to the bottom than the top. This will tend to give greater lead at the bottom and later cut-off on the up stroke as compared with the top and the down stroke. The attempt to produce any considerable modification in this manner will produce, however, undesirable disturbance in the points of steam opening, and reduction in the width of port opening on top. At best the modification of the lap must be counted in as only one among various means available for bringing about such a final combination of valve events and items of port opening as may be considered most satisfactory for the case in hand.

We shall now turn to a more detailed examination of the subject of block slip, and its relation to the various proportions and configurations of the gear.

In Fig. 29 we assume an open-rod gear fitted to an engine with back columns on the left-hand side (upper side in the figure), and hence with crank turning in the direction of the arrow. With an outside valve also, this gives P as the position of the go-ahead eccentric, while A will be the go-ahead end of the link. The point of attachment of the bridle or suspension rod is either at R or A, and so far as possibilities go, the rods may lie either to the right or left, locating in the one case the rock shaft on the front columns and in the other on the back. This will give four possible cases as follows:

(1) Point R in middle of link with rock-shaft on front columns.

(2) Point R in middle of link with rock-shaft on back columns.

(3) Point R at go-ahead end of link with rock-shaft on front columns.

(4) Point R at go-ahead end of link with rock-shaft on back columns.

We may note at once that case (4) is not met with in usual practice on account of the short length of bridle rod which would result, and the consequent increase in block slip. This case may therefore be disposed of without further notice.

In Fig. 31, series B, a series of curves is given showing paths of *A* for case (1). The point of attachment *R* is taken in each case on the center line of the arc of the link, while the various curves result from shifting the position of the center *S* parallel to the engine center line. The paths lettered from *a* to *f* show the result of a continuous rise* in the center *s*, producing the transformations throughout the series, and showing the possibilities regarding block slip for these various locations of *S* with the other items fixed as specified. It will be clear in any given case that the total amount of slip will be given by the total breadth of the curve or path followed by *A*, estimated at right angles to the line of motion of the valve rod, which in each case is denoted by a vertical line drawn through the curve. This line furthermore divides the total excursion of the point *A* into equal parts on either side, and it may thus be taken as a convenient reference or center line relative to which the slip of the block on either side of its mean position may be noted.

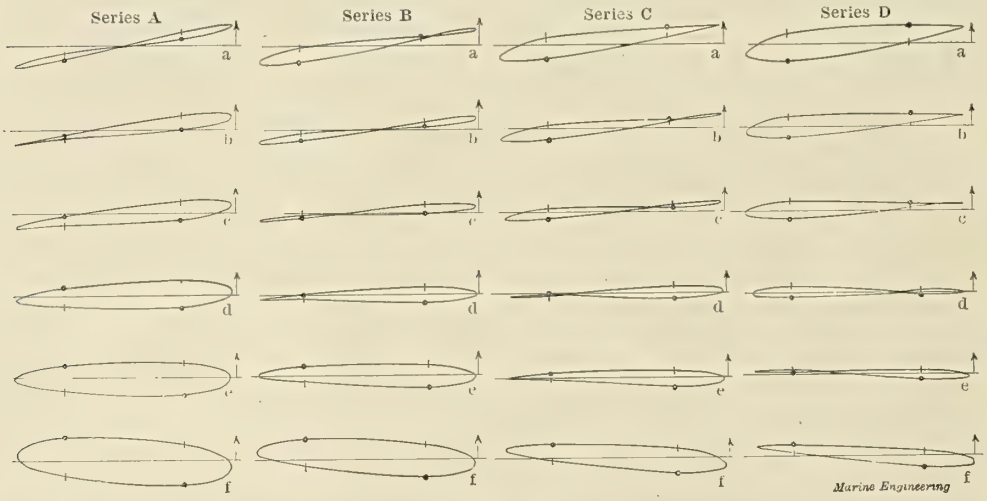


FIG. 31.—OVALS SHOWING PATHS OF GO-AHEAD END OF LINK, CASE (1). IN EACH CASE THE UPPER END OF THE PATH IS DENOTED BY THE SMALL ARROW, WHICH POINTS TOWARD THE BACKING END OF THE LINK.

In Fig. 32, series B, a similar series of curves is given showing paths of *A* for case (2), the point *R* being likewise taken on the center line of the arc of the link, while the various paths, as before, result from shifting the position of *S* in the vertical direction. As in Fig. 31, the curves lettered from *a* to *f* show the result of a continuous rise in the center *S*, and the resulting changes of form show the varying values of the slip as determining by the breadths of the paths, and relative to the center line drawn in each case.

In case (3) the paths of *P* must necessarily be arcs of circles more or less inclined to the line of motion of the valve according to the location of *S*. Such a series is shown in Fig. 33, not, however, taken from a model, and somewhat exaggerated in curvature to better bring out the tendencies in this case.

It will next be of interest to note the results of a change in the location of the point of attachment *R* relative to the center line of the arc of the link. In Fig. 31 are given four series of curves for case (1), and in Fig. 32 four like series for case (2). The series lettered *A, B, C, D*, result respectively from four locations of the point of attachment *R*, *A* just within on the concave side, *B* on the arc, *C* just without on the convex side, and *D* still farther outside. The various paths within each series correspond to a succession of locations of the center *S*, from below upward in each case, corresponding to progression from left to right in the diagram.

A comparison between similar paths in these various series will then serve to show the effect of shifting the point *R* across the line of the link in the manner stated.

*In the present discussion the terms signifying direction have reference to a vertical engine, and not to diagrams like Fig. 29, which are turned down for economy of space.

ABSTRACTS FROM RULES AND REGULATIONS OF THE AMERICAN BUREAU OF SHIPPING FOR OIL-TANK VESSELS ONLY.

EDITION OF 1904.

ARRANGED BY HARRISON S. TAFT.

1. General.

Herein rules, etc., are to cover only that part of ships devoted to an oil or other liquid cargo.

2. Metacentric Height.

Said height should be from 15 to 21 inches for steamers; but from 30 to 36 inches for sailing vessels.

3. Length of Holds, etc.

The fore-and-aft length of an oil tank should not exceed 32 feet.

The oil should go to the skin of the ship. When double bottoms are allowed for water ballast purposes means must be provided for a thorough ventilation of the same. There must be an oil-tight center line longitudinal bulkhead extending to the top of the expansion trunk when said trunk is worked at the center line; otherwise to the "crown deck."

4. Cofferdams.

A cofferdam should be fitted at each end of the spaces devoted to an oil cargo. Fore and aft they should be two frame spaces wide. They are to extend from the keel to at least the deck forming the crown of the oil tanks.

5. Expansion Tanks.

At least one expansion tank is to be fitted to each hold or tank. The capacity of said tanks to be at least 7 percent of the capacity of the hold to which they are fitted for an oil cargo; but 4 percent for a water cargo.

6. Test Pressure Head.

The pressure head for testing tanks should be at least 4 1-2 feet above the top of the expansion tanks; or 12 feet above the "crown deck."

7. Middle Line Keelson.

1. Said keelson is to be 2 pounds heavier than specified for floor plates of ordinary vessels. It should extend above top of floors a sufficient height so as to be embraced by the two continuous fore-and-aft angles on top of the horizontal plate stringers on top of floors (see paragraph No. 2), and to allow of a double riveted lap to the center-line bulkhead plating, with sufficient room for caulking. The lower edge of said keelson is to be secured to a flat plate keel as in ordinary ships.

2. Horizontal plate stringers, each in width at least 3-4 the depth of floors at the center line, and equal to the weight of the center-line keelson, are to be worked on top of floors on each side of the center keelson, and be secured to same by the above fore-and-aft continuous angles.

3. The center-line keelson and floor stringers are to maintain their midship scantling for 1-2 length amidships.

8. Frames.

1. Frames may be of the same flange size as for ordinary vessels, but 10 percent heavier; or of the compound type. They should be cut 8 inches short from the center-line keelson, but they must have a 3-rivet connection to a flat plate keel. Oil- or water-tight transverse bulkheads are to have a single frame bar. Said bar is to be of sufficient flange width to provide for a double row of zigzag rivets to both the shell and bulkhead plating.

2. With a center-line expansion tank all frames are to be cut at the stringer plate of the "crown deck"; said frames being bracketed above and below to said stringer plate with bracket plates 10 percent thicker than the flanges of the frame bars. The brackets on the lower side of said stringer shall have at least a 5-rivet connection to the frames, with a similar connection to the deck beams or stringer plate when single riveted; but a 10-rivet connection when double riveting is called for. The brackets on top of said stringer plates may have a flange or clip connection to the stringer, with same riveting as for the brackets on the under side of said stringer. For a clip connection, the clip flanges must be equal to the thickness of the bracket plate; for a flange connection the weight of said brackets must be 2 pounds heavier than for brackets with a clip connection to said stringer. The width of the clip flanges or of the flange on the plate must be wide enough to meet the size and style of riveting required by said stringer plate. The diagonal edge of the above brackets is to have a 3-inch by 3-inch clip or a 3-inch flange.

3. With a side expansion tank the frames are to extend in one piece to the upper deck; the main deck stringer plate being scored around the frames and fitted with clips to the shell plating. There shall be an angle bar at the face of the frames as for a regular side stringer.

4. The scantling of the frames above the "crown deck" is to be the same as that below said deck on corresponding frames.

9. Floor Plates.

1. The floor plates are to be of the same size and description as the floors in the machinery space of ordinary vessels; or they may be worked straight on top if bracket plates are fitted at the bilge.

2. Floors straight on top are to be 10 percent deeper, or of the same depth and 10 percent heavier than the floors of ordinary vessels; but in no case are they to be less than the lower third of bulkhead plating. The bilge

brackets are to be of the same weight as the floors and are to have a 6-rivet connection to the floors and the frames for single riveting; but a 12-rivet connection for double zigzag riveting. Said double riveting is to be used when the second number is or above 25,000. Said bilge brackets are to be omitted on web frames.

3. In lieu of a stiffening bar on the top edge of the bilge brackets, said edge may be flanged. When the second number is under 19,000 said flange is to be 3 inches wide; when said number is 19,000 and under 52,000, said flange is to be 3 1-2 inches wide; when said number is 52,000 or over, said flange is to be 4 inches wide. Angle-bar stiffeners are to have corresponding flanges and are to equal the thickness of the bracket plates.

10. Reverse Bars.

1. With channel, Z-bar, or bulb angle frames, the reverse bars are to be of the same scantling as for an ordinary vessel. They shall extend from the center line to the outer end of the floor plates on every floor, having reverse clips of the same size at all floor keelsons.

2. With compound frames they are to be of specified size, and shall extend to the upper deck on every frame, being cut at the "crown-deck" stringer.

3. With angle-bar frames, they are to extend to the "crown deck" on every frame.

4. On web frames they are to be worked continuous from the center line to the first side stringer above the floors, and intercostally between the remaining side stringers and the "crown deck." Doubling clips are to be worked at the diamond plate connections of all side stringers.

5. When the deep-frame construction is adopted the reverse bars are to have a 2 1-2-inch lap to the frames when the first number is under 60; a 3-inch lap when said number is 60 but under 100; and a 3 1-2-inch lap when said number is 100 or above.

6. When the shorter flange of the reverse bar is riveted to the top of floors, it is to have a single row of rivets to same, spaced 6 diameters apart; but when the longest flange is riveted to the floors a double row of zigzag rivets, spaced 7 1-2 diameters apart, is to be used.

11. Floor and Bilge Keelsons. Side Stringers.

1. FLOOR SIDE KEELSONS.

(*a*) Vessels of less than 30-foot beam are to have one intercostal floor keelson on each side of the center line. The intercostal plates are not to be less than 9-10 of the thickness of the floor plates, and they are to extend above top of floors, so as to be embraced and riveted between two fore-and-aft continuous keelson angles. They are to have single clips to the floors and shell plating. They shall extend from peak to peak with the above fore-and-aft angle bars worked continuously through all transverse bulkheads. The intercostal plates are to extend forward and aft as far as possible. When two lines of keelson bars approach each other to within 3 1-2 feet, one line of said bars may terminate at said point.

(*b*) For vessels of 30-foot beam or over see Table 1.

2. BILGE KEELSONS.

(*a*) There shall be a double angle-bar bilge keelson at the lower turn of the bilge. A single bulb angle may be used if its sectional area is at least 85 percent of the combined area of the two above angle bars. Said keelson is to be worked continuous from peak to peak.

(*b*) When the beam is 30 and under 36 feet, or 42 and under 48 feet, or 54 and under 80 feet, the above bilge keelson is to have a plate stringer equal to the weight of the floor plates, and in width at least 1-2 the depth of the floors at center line, worked under said keelson bars and riveted to the reverse bars and clips on top of floors (See Table 1). If said stringer plate is cut at a bulkhead, bracket plates equal to the weight of the stringer plate are to be fitted to both sides of the stringer plate on each side of the bulkhead. The width of the bracket plates on the bulkheads is to be at least 3-4 the depth of the floors at the center line, and on the stringer plate one frame space. The brackets are to be double clipped to the bulkheads with both flanges double riveted, and they shall be double riveted to the stringer plate. Said stringer plate is to extend for 3-4 length amidships. In lieu of said plate stringer, a bulb plate of specified depth may be riveted between the two keelson bars.

(*c*) Water-tight collars are to be fitted on both sides of all bulkheads at the floor and the bilge keelson bars, with both flanges double riveted.

TABLE 1.—FLOOR AND BILGE KEELSONS.

Maximum Beam.	Number of Floor Keelsons.	Type of Bilge Keelsons.
Under 30 ft.	1	D A
30 ft. " 36 "	1	D A + H P S
36 " " 42 "	2	D A
42 " " 48 "	2	D A + H P S
48 " " 54 "	3	D A
54 " " 60 "	3	D A + H P S
60 " " 70 "	4	D A + H P S
70 " " 80 "	4	D A + H P S

D A = double angle bars.
H P S = horizontal plate stringer under keelson angle bars.

3. SIDE STRINGERS.

(*a*) They are to equal the thickness of the web frames and be worked continuous from bulkhead to bulkhead. They shall be spaced not more than 5 1-2 feet apart, or as stated in Table 2.

(*b*) When said stringers are cut at a bulkhead, bracket plates equal in weight to the stringer plate, extending on the bulkhead from said stringer at least 2 1-2 times the width of said stringer plate and the same distance on the

TABLE 2.—NUMBER OF SIDE STRINGERS.

Height from the Top of Floors to the Stringer Plate of the "Crown Deck."	Number of Stringers.
Not over 11 feet.	1
Over 11 ft., but not over 16½ ft.	2
" 16½ " " " " 22 "	3
" 22 " " " " 27½ "	4
" 27½ "	Hold Beams.

stringer plate from the bulkhead, are to be worked on both sides of the bulkhead. Said brackets are to be connectioned to the bulkhead by a double-riveted flange or by double-riveted angle bars, the thickness of the angle flanges being at least 3-4 the thickness of the bracket plates. The brackets are to be double riveted to the stringer plate and have their diagonal edge flanged.

(*c*) When the plate stringers are continued through a bulkhead, double angle bar collars double riveted in both flanges are to be fitted on both sides of said bulkhead, the flanges being at least 3-4 of the thickness of the stringer plate. Bracket plates, 1 1-2 times the width of the stringer plate on each side with a flanged diagonal edge, are to be worked on each side of the bulkhead; the connection of said brackets to the bulkheads and side stringers being the same as when the plate stringer is cut at a bulkhead.

(*d*) Side stringers are to be clipped to the shell, with the continuous angle bar on the face of the frames as for an ordinary vessel.

(*e*) When the second number is not over 19,000, a single face bar is to be fitted to the inner edge of the stringer. When said number is 19,000 and over, the face bar is to be worked double; the longest flange being vertical in either case.

(*f*) Knee brackets, of at least 85 percent of the weight of the plate stringer, are to be fitted at every frame extending the full width of the stringer plate, with the same width on the frames; but the latter need not exceed 24 inches.

12. Floor Clips.

Reverse frame clips are to be fitted at all keelsons, plate side stringers, and continuous fore-and-aft angle bars. When bulb angle frames are used said clips are to have a 2-rivet connection to the stringer angles.

13. Web Frames.

1. Web Frames are to equal the weight of the floor to which they are attached. They are to be worked on every fourth frame throughout all tanks and machinery space, having at least a single face bar. Said webs are to extend from the floor plates to the "crown deck." They are to be cut at the side stringers, having clips to same with double riveting in both flanges. Besides being riveted to their respective deck beam they are to have bracket plates to said beam of the same weight as the web plates, with at least an 8-rivet connection to said beam and web respectively. When the width of the frame flange permits, the web should have a double zigzag riveted connection to said frame. Diamond plates, equal to the thickness of the web frames, are to be worked at all side stringers. Said plates are to have at least 3 rivets to each face bar on the web frame above and below the stringer angle bar, and 5 rivets on each side of the web face angles to each face bar on the stringers.

2. Between decks, on the bulkhead frames, web plates with a reverse face bar and a double-riveted clip to the "crown-deck" stringer are to be worked throughout the tank space.

14. Deck Beams and Knee Brackets.

1. Beams at web frames; beams at ends of hatchways or other deck openings; hatchway half-beams, etc., are to be deeper than the other beams throughout the oil-tank space.

2. The beams of the "crown deck" are to be spaced at every frame throughout the oil-tank space. They are to extend from the frames to the center-line bulkhead, except in way of hatches where half-beams are to be used. Said half-beams shall have a 4-rivet double-clip connection to the fore-and-afters at said hatches.

3. The upper deck beams should be cut at the expansion tank trunk and be securely fastened to the same with bracket plates and clips, so as to maintain the continuity of strength.

4. Plate knees are to be 2 pounds thicker than the beams, and are to be in depth and width three times the depth of the beam to which they are secured. They should be flanged on the diagonal edge.

5. All hold beams are to be spaced at every web frame. They are to be located so as to connect to the second side stringer down from the "crown deck" with horizontal bracket plates, having at least a 10-rivet connection to the side stringer on each side of the web frame face bar. They are also to have a vertical bracket to the web frame with at least a 6-rivet connection to both the web and the beam. Similar horizontal and vertical brackets are to be worked at the center-line bulkhead. Bracket plates are to equal the weight of the side stringers and web plates.

15. Deck Stringers and Deck Plating.

1. The deck stringer of the "crown deck" is to be as specified for ordinary vessels; except they are to be 2 pounds heavier when the second number is under 25,000; 3 pounds heavier when the second number is 25,000 and under 90,000, and 4 pounds heavier when the second number is 90,000 and over. Said stringer gunwale bars are to be double zigzag riveted.

2. When the expansion tank is located at the sides of the ship, said "crown-deck" stringer is to be secured

by intercostal clips to the shell plating and have a similar angle bar on the face of the frames throughout said tanks as for a regular side stringer. The above face bar is to have a welded knee at the ends of each tank and be worked across the full width of the stringer plate; being riveted to it and through the bulkhead plating to the bounding angle bar on the other side of the expansion-tank plating.

3. The deck plating of the "crown deck" is to be 2 pounds heavier than that of the upper deck of ordinary vessels when the second number is under 36,000; 1 pound heavier when said number is 36,000 and under 72,000; but of the same weight when said number is 72,000 or above. When said deck plating is flanged to connect with the expansion tank plating, said flange plate is to be made 2 pounds heavier than otherwise.

4. When the expansion trunks are worked continuous the strake of the deck plating next to said trunk is to be made heavier as specified.

5. The liners under the upper strakes are to be cut short so as to allow for ventilation.

16. Stanchions.

1. When the beam of a vessel is 42 and under 60 feet, a row of quarter-stanchions is to be worked on each side of the center line. When said beam is 60 feet or above, two rows of stanchions are to be worked on each side.

2. When the length of a stanchion is less than 15 feet, it shall have a 2-rivet connection at each end. When of 15 feet or over, it shall have a 3-rivet connection to the beams, but a 4-rivet connection to floors, with a reverse clip at the latter. Said clip to have a 4-rivet connection to the floors.

3. Stanchions may be worked on the web frame beams only, if a plate girder fitted with double continuous angles on the lower edge is scored around the deck beams; being clipped to the deck plate and bulkheads; and having clips at the heavy deck beams to the said fore-and-aft bottom angles. The head of the stanchion is to be secured to said girder. Said stanchions are to rest upon a flat diamond floor plate, equal to the weight of, and embracing three floors. Said diamond plate is to have a 3-rivet connection to the floors at its ends, but a 6-rivet connection to the floor at its middle when the length of the stanchion does not exceed 15 feet. When the length of a stanchion is over 15 feet, 4 rivets are to be used at the end floors and 8 rivets at the middle floor. But if said stanchion comes on a side keelson said diamond plate may be omitted.

17. Bulkheads.

A. Transverse Bulkheads.

1. Said bulkheads are to be spaced not more than 32 feet apart.

2. Bulkhead frame bars are to be worked single, but they are to have a double row of zigzag rivets in each flange. Similar angle bars are to be worked at the junction of a transverse with the continuous longitudinal center-line bulkhead; but they are to be worked on opposite sides, so as to avoid three-ply riveting.

3. The shell liners are to be wide enough to allow of a double row of zigzag rivets on each side of the bulkhead bar.

4. (*a*) Vertical stiffeners of extra weight are to be spaced not more than 4 feet apart, being securely riveted to the floor plates. They are to have top and bottom brackets, with a 6-rivet connection to the stiffeners, but a 5-rivet connection to the deck plating and floor plates. The bracket plates are to be 2 pounds heavier than the lower part of the bulkhead plating. Intermediate stiffeners of regular size are to be worked between the above stiffeners.

(*b*) When the extreme breadth of a bulkhead is 40 but under 55 feet, one web plate, equal to the weight of a regular web frame, is to be worked at the stiffener at the middle of the half breadth. Said webs are to extend from the shell plating to the "crown deck," being one frame space wide for the lower third with a taper for the upper 2-3 to a width of 1-2 of a frame space at the deck. They are to have at least a single face bar equal to the reverse bars of an ordinary ship. Bracket plates, equal to the lower part of the bulkhead plating, are to be fitted to said web in way of the horizontal stiffeners, securing said web to the bulkhead plating in way of said stiffeners. When the breadth is 55 feet or over, two such webs are to be worked on each side of the center line.

5. Horizontal plate stiffeners are to be worked on the opposite side to the vertical stiffeners in range of all the side stringers, to which they are to be bracketed. Said plate stiffeners are to be connected to the bulkhead plating with a single bar, and have a face bar of similar size. When said stiffener plate is over 10 inches in width, bracket plates of the same weight as the stiffener plate are to be worked in way of the heavy vertical stiffeners with a flange or clip connection to the bulkhead plating; having a 5-rivet connection when the second number is under 19,000, but 7 rivets when said number is or over 19,000.

6. The plating and stiffeners of the cofferdams are to be the same as above. The vertical stiffeners should be worked on the inside, with bracket plates equal to lower plating of bulkheads worked on all the heavy stiffeners and in way of all the horizontal stiffeners. Said bracket plates are to have at least a 5-rivet connection on each side.

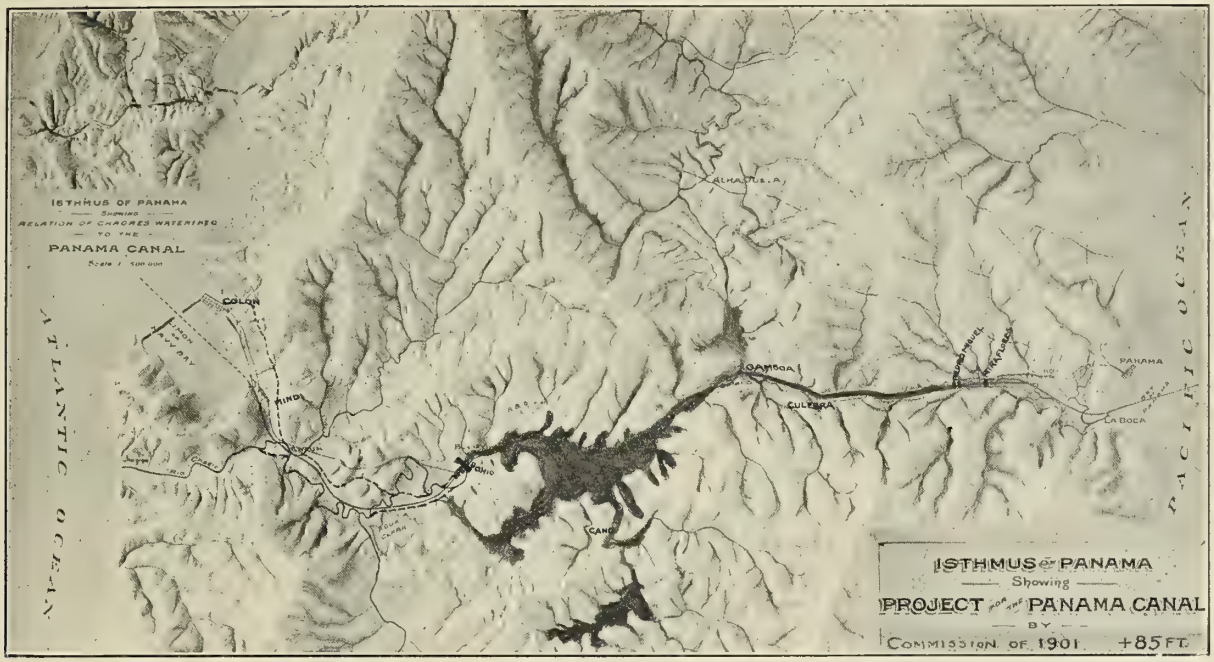
B. Longitudinal Bulkheads.

1. The plating is to equal that of the transverse bulkheads, except for the top plating to the "crown deck," which is to equal the lower strake of the bulkhead plating. They are to extend to the "crown deck" or to the top of the expansion tank.

2. When worked continuous the longitudinal bulkhead is to be connected to the transverse bulkheads as stated above. When worked intercostally they are to have a single-riveted double angle-bar connection to the athwartship bulkheads. The flanges of said single bars are to equal the thickness of the lower plating.

3. Vertical stiffeners are to be spaced as per frame spacing. Said stiffeners are to be bracketed to the deck beams and to the floor stringer plate. Said floor stringer brackets are to equal the weight of the lower part of the bulkhead plating. They are to have a 5-rivet flange or clip connection to the floor stringer and stiffener angle when the second number is under 19,000, but a 7-rivet connection when said number is 19,000 or over.

(To be continued.)

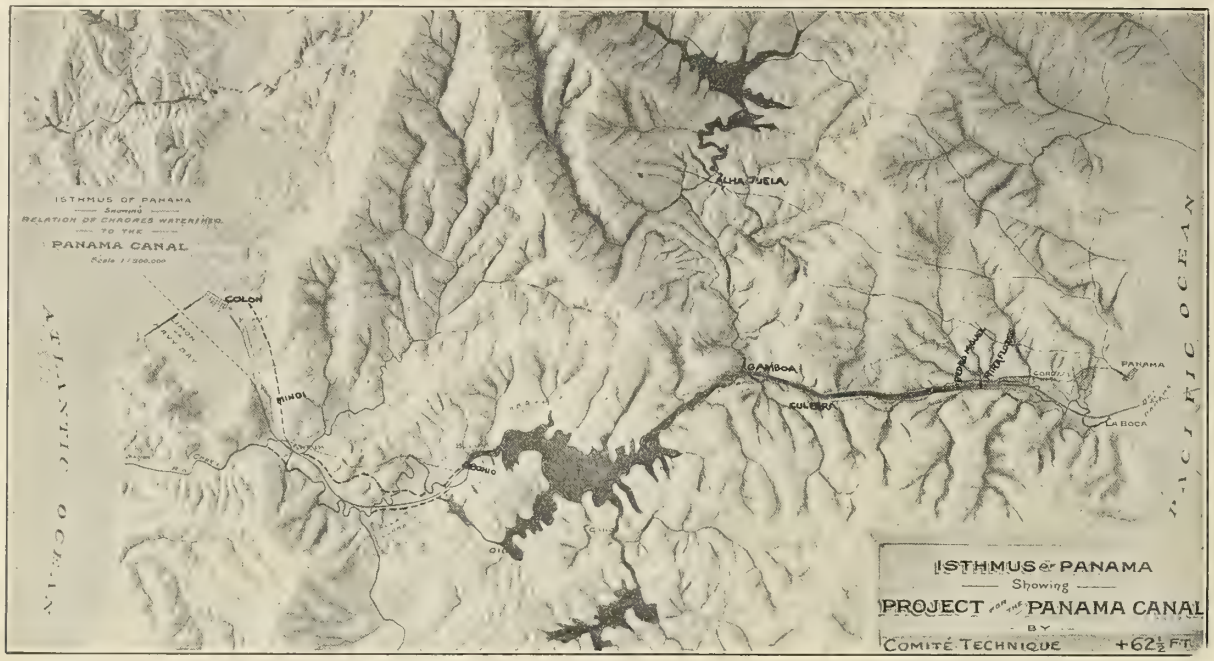


THE PANAMA CANAL.

The six canals, whose plans are given in the accompanying cuts, record the story of one of the greatest public works undertaken in all history. When in 1881 DeLesseps actually began operations they were for a sea-level channel some 29 feet deep. Ships were to traverse it in 1889 and its cost was to reach $127,500,000. When, however, DeLesseps found himself up against the pestilent morasses of the Panama coast and the jungled swamps, the rocky Culebra and the torrent of the Chagres, when 6,000 of his own countrymen lay buried down by the two seas, he had measured forces with Nature truly, and he fell back baffled. After ten years had passed $257,000,000 had been spent and less than two-fifths of his small canal was finished. It was the realization of all that DeLesseps' failure implied, which moved our commission of 1901 to reject a sea-level totally. For a 35-foot sea-level canal it estimated $240,000,000 and a period of at least twenty years.

It is perhaps only natural that, appreciating the combination of natural forces which had so overwhelmed the French with disaster, this first American commission should seek safety above all things and should wish to escape just as far as humanly was possible the perils which proved so fateful to the sea level. It recommended, therefore, a canal at the very other extreme—the 85-foot project.

This is the plan upon whose estimates the Panama investments were bought, and the $145,000,000 appropriated by our Congress. It included a series of big lift locks, consuming much time for their passing. But the higher the channel the less the menace of the dreaded Chagres, and the less chance for duplicating the expenditure of the French, and having no available canal at the end. The 85-foot, like the Comité Technique and the 30-foot plans, all contemplated the same general treatment of the river floods. A very high dam at Bohio must hold back a deep lake several miles in area. Now, Bohio is situated where the river breaks through the median ridge. It is self-evident that an accident to this dam must precipitate the impounded lake upon the lower country in a disaster beside which that of Johnstown would have been slight. The dread of the dam is the reiterated refrain of every report heretofore made on the Panama canal. In a country where earthquakes are far from uncommon, a breach in the Bohio dam was an ever-present menace. Then, too, the water from the sudden tremendous floods must be received in this lake, and its overflow



was the next crucial problem. It could not be left to run down into the canal, so diversion channels, larger than the canal itself, had to be devised. (Shown on maps in dotted lines.)

In the late borings has come a *coup-de-grace* for the 85-foot project. Upon the site of the Bohio there have been found no sure foundations for the very high masonry core-dam it necessitated. This has struck at the very vitals of its possible construction. So it is regarded as abandoned.

The 30-foot has been only vaguely outlined. No detailed estimates have been given out.

The "new sea level," reviewed and recommended by three members of the last commission and the chief-engineer, advocates a 35-foot channel, and gives the cost for this of $230,500,000. The new Cunarders will draw 37 feet. It is evident, therefore, that a 35-foot depth is quite inadequate for present let alone prospective, needs.

The 62.5-foot canal of the Comité Technique has the same elements, though lesser in degree, of high dams, deep impounded high-level lakes, an unnecessarily tortuous channel, the retention of the death-dealing swamps, an approach hard to negotiate, and with superfluous curves. Its estimate also exceeds the appropriation of Congress.

The remaining canal, that designed by Mr. Lindon W. Bates, is seen to be much more direct and much less curved. From approach to approach it is nearly three sea miles the shorter. It proposes a different entrance from all the others, and on the Pacific end eliminates 66 degrees of curvature. At both termini it creates new town sites by utilizing the material dug from the Culebra. It proposes end locks which shall hold the canal at a level from 20 to 25 feet. These locks are located in rock. Adjacent to one of them the French made a rock cut of 70 feet, so the foundations are beyond question.

Continuous with these locks are built barrages which inclose two terminal lakes, Lakes Chagres and Panama. These terminal locks and lakes are the solution of the first group of problems. The locks being at the end where ships must halt for taking out papers, etc., are passed incidentally without actual loss of time, as is necessitated by the interior lock systems of all the other projects. The lakes give in Project "A" nearly 18 miles, and in Project "B" nearly 26 miles of free, open navigation, in which a vessel can make good speed. This allows a great gain in time over the restricted speed of a tortuous, narrow channel.

The next great feature of the new plan is that these terminal

lakes submerge all the fateful swamps which have made the name of Panama almost a synonym for disease and death. This sanitary achievement is winning for the Bates canal the alliance of all the medical journals. It is momentous not alone for the period of construction but for all after time to the residents of the terminal cities and to all the shipping which shall traverse the canal.

The lakes furnish also the means of reducing vastly the unit cost, since they are located where virtually no further work is entailed in their creation save the end dykes. They enable much of the Culebra cut excavated material to be transported by floating plant, which is much the cheapest method of treatment.

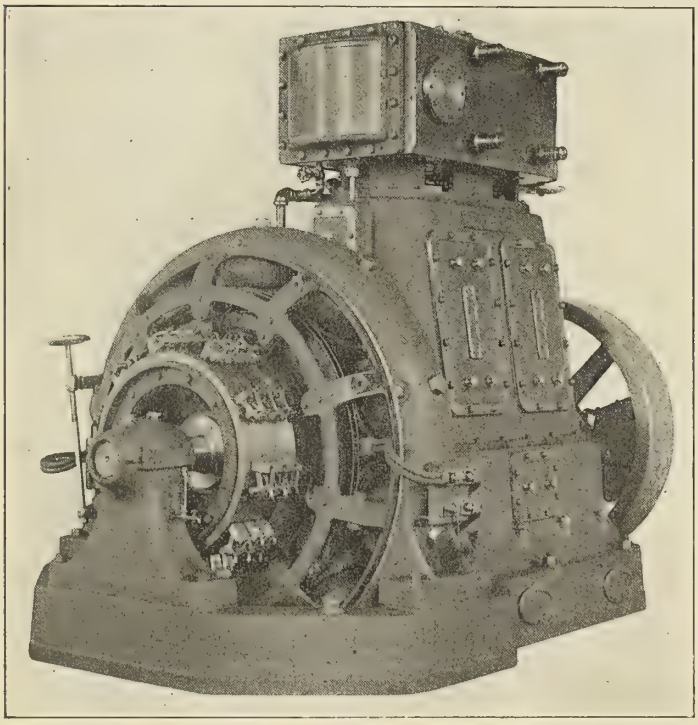
The other historical problem, "the Chagres," is attacked from an entirely new point of view by Mr. Bates. Where the commissions have proposed high dams, he substitutes low dams and barrages, such as can with unquestioned safety be built upon the foundations, as borings have actually revealed them. For the head waters of the Chagres he provides catchment basins which are to be empty, except when the sudden torrential floods are temporarily impounded in them. At Alhajuela and Gamboa are built under-sluice dams, by means of which the volume of the river is controlled and its water is fed down to a triangular basin, the Obispo Basin. A gradually expanding ratio is applied to regulating the current toward the lakes. In the Obispo Basin the Chagres is divided into two halves, and each half is led down through the canal itself to either ocean. The halving of the river after its volume has been controlled to the rate of absolute safety takes away the "Chagres terror," which has haunted every one of the thirty odd engineers who have reported upon it. It is stated upon the authority of one long a resident and student of Isthmian problems, that every solution Mr. Bates has presented is feasible and practical.

Owing to the elimination in all of 114 degrees from the canal curvature, the cutting off of nearly three miles in length, the high speed of its lake navigation, and the fact that no time is lost at his locks, Mr. Bates is able to show a reduction of five hours, or nearly 50 percent, in the time of passing through his canal, over the time demanded by every other. Owing to the diminution of unit price made possible by the terminal lake methods of operation and the deduction of some 18 or 26 miles of construction, this canal can be brought within the $145,000,000 appropriation. For the same general reasons it can be completed in eight years instead of in the 10 to 20 years demanded by the other schemes.

ENGINEERING SPECIALTIES.

A New Marine Generating Set.

The generating set which we illustrate herewith is being built by the B. F. Sturtevant Company, of Hyde Park, Mass., in five sizes ranging from 17 1-2 to 100 Kw., the latter being designed to operate at 350 revolutions per minute under a steam pressure of 150 pounds per square inch. The total weight of this set is 22,000 pounds. The motive power consists of a cross-compound



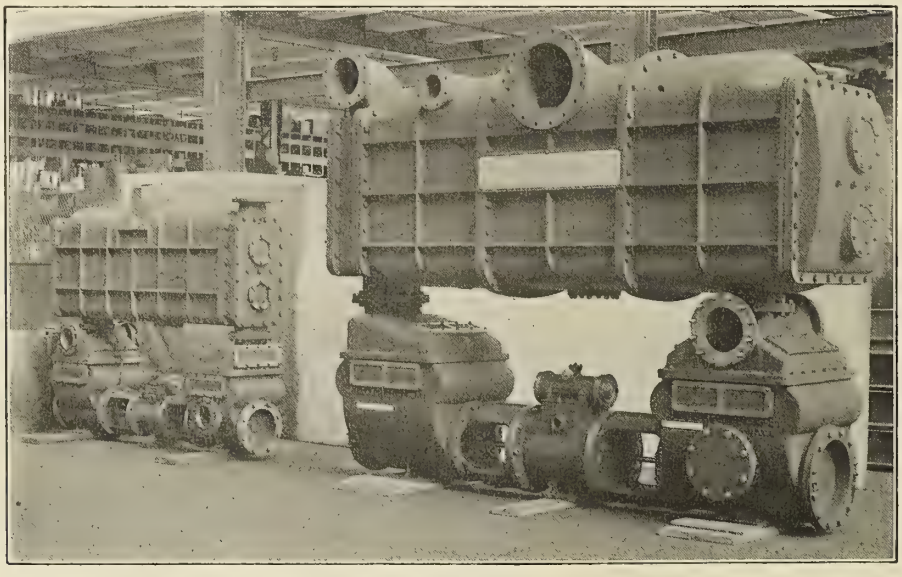
engine, with cylinders measuring 10 and 18 inches in diameter, and a stroke of 10 inches, and operating on a steam pressure of 150 pounds per square inch. The steam and exhaust pipes measure respectively 4 and 6 inches, the crank shaft has a diameter of 5 1-2 inches, and the crank pin 5 1-2 inches by 6 1-2 inches long. The multipolar generator has been designed so that it will be capable of carrying for short spaces of time as much as 50 percent overload, while it is to be capable of operating for long periods at 25 percent overload, without undue heating. It is, moreover, so designed that the jump in the voltage when full load is suddenly thrown on or off shall not exceed 15 percent. The Rites governor is expected to hold the speed within 1 1-2 percent of normal when full load is thrown on or off.

The smaller sizes have capacities respectively of 17.5, 25, 35, and 50 Kw.; their weights complete being 5,000, 7,300, 10,000, and 14,000 pounds. The cylinder dimensions are: 6 1-4 and 10 1-2 by 6 1-4; 7 and 12 by 7; 8 and 14 by 8; and 8 and 14 by 8; the steam pressure in the last case being 150 pounds, and in the other cases 100 pounds per square inch. Although originally designed and built for the United States Navy Department, they are now being turned out for the general trade and are constructed and tested in as thorough a manner as when built under government orders.

A New Surface Condenser.

Placed on the market by the W. H. Blake Steam Pump Company, of Hyde Park, Mass., the two condensers shown in the illustration, and intended for marine use, have tubes of Muntz metal held in the tube sheets in such a manner as to allow free expansion. The steam enters at the top, and rapid condensation is secured by a liberal supply of baffle plates which force the steam into close contact with the exterior of all the pipes through which the cooling water flows. The pump of the large condenser illustrated has a 16-inch steam cylinder, 18-inch water cylinder, and 18-inch air cylinder, with a common stroke of 18 inches, the capacity being 25,000 pounds of steam per hour with 2,000 square feet of condensing surface. The suction and discharge openings in the air cylinder measure respectively 12 and 10 inches and in the circulating cylinder 12 inches. The small condenser has cylinders measuring 8 inches for steam and 12 for air and water, with a 12-inch stroke. With 450 square feet of condensing surface, the capacity is rated at 14,400 pounds of steam per hour, suction and discharge pipes being 8 and 6 inches for air and 8 inches for water.

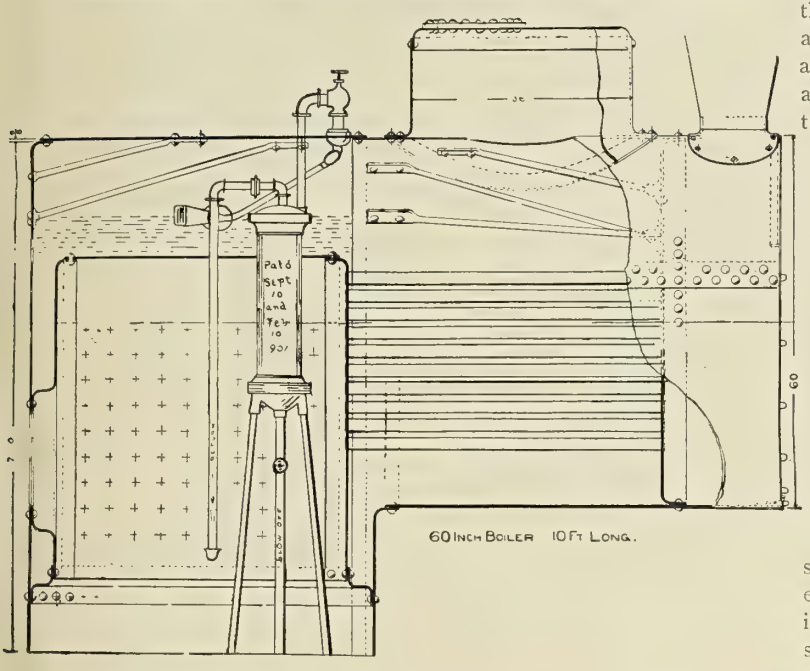
Absolute certainty of action is insured, for the valve mechanism is actuated solely by direct boiler pressure, and is not dependent upon an adjustable arrangement of levers and rods connected to the piston. With the latter arrangement, which is not employed upon the pumps, there are certain points in the stroke where the valves are not directly controlled, but momentum is relied upon to reverse the valve gear. These pumps are composition fitted throughout, both the water and air cylinders are lined, and a Tobin bronze piston rod is used.



Buckeye Boiler Skimmer.

The illustration shows one of these skimmers in position on a horizontal tubular boiler. It may be said in general that the skimmer consists of a wide-mouthed funnel suspended by means of a tubular arm and floated at the surface of the water in the boiler by means of two copper floats tested to a pressure

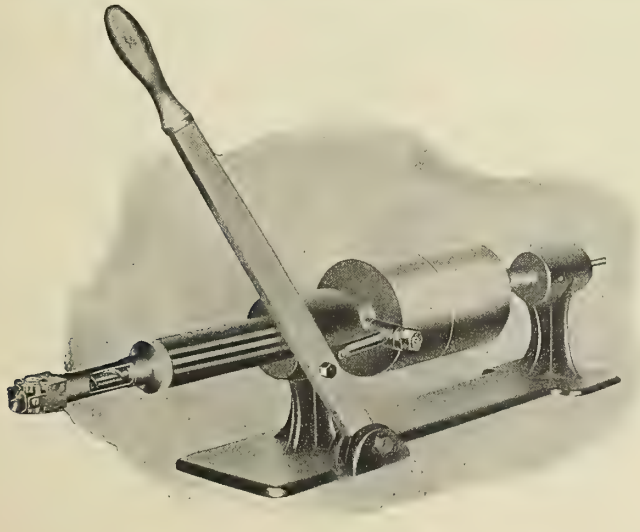
of 500 pounds per square inch, and guaranteed against either collapsing or leakage. This skimmer is stated to be especially adapted to use in very dirty or muddy water, such as is found on many of our Western rivers, and is guaranteed to keep a boiler



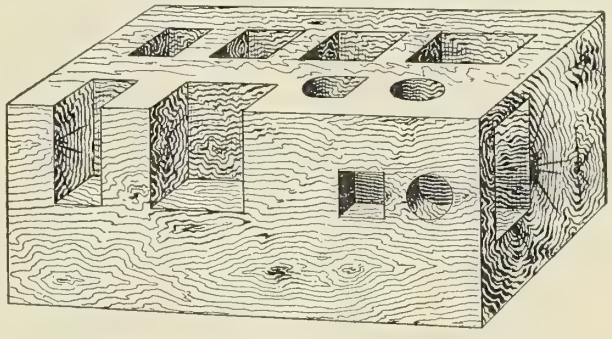
absolutely clean from such impurities when properly installed and operated. At the upper end of the arm is a flexible joint, which is both water-tight, and of such ease of operation that the floats may readily maintain the elevation of funnel desired. This implement is placed on the market by the Buckeye Boiler Skimmer Company, of Toledo, Ohio.

A New Square Auger.

An extremely interesting device, in the shape of an auger for boring square holes, is being placed upon the market by the American Pressed Steel Company, of Philadelphia. This tool should be a great labor-saving device for shipyard use on account of the rapidity with which it bores or cuts a mortise. Much time is very often lost in a shipyard in squaring out the holes for countersinking square-headed bolts, by hand, in the timber for ceiling battens or in hatch covers; and the cost of a machine and bit might easily be saved on one job. One of the machines has cut a mortise in knotty maple, one inch wide, four inches deep, and six inches long, in forty-five seconds. This mortise did not require trimming at all, the corners were accurately and cleanly cut, and although the auger was run at a high speed there was not the slightest tendency to split the wood or overheat the bit.



Although the manufacturers make this auger to fit any boring machine, for exceptionally rapid production of work they recommend their special mortising machine. All points of friction are lubricated continually through a protected channel, thus enabling the bit to be run very rapidly. Another feature in this machine is an arrangement by which the bit can be set to cut at any desired angle on the timber, or endwise of the grain. The machine is portable, as it weighs only about seventy pounds, and can be set up on a trestle or bolted to a bench. The various cutting parts of the

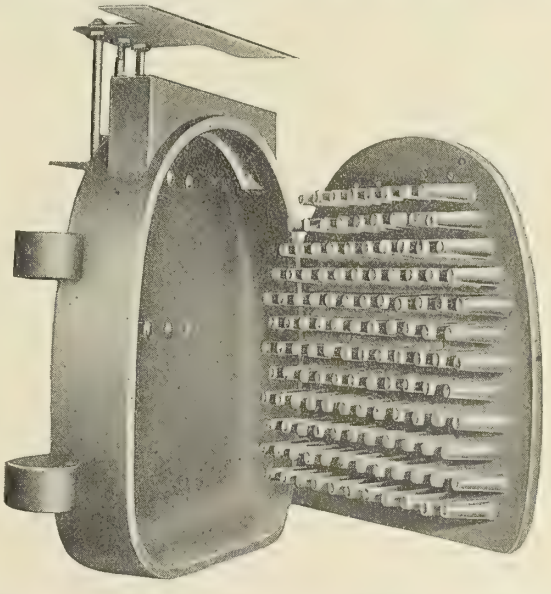


EXAMPLES OF THE WORK OF THE PEARL SQUARE AUGER.

square auger can be renewed, and are interchangeable. They are easily sharpened, and are made of the best quality tool steel. It is claimed for this tool that it is the only auger that makes a square hole or mortise with a rotary motion and bores out the wood at the same operation, leaving clean-cut corners; the only tool with which it is possible to make a mortise at any desired angle on the timber, or endwise of the grain; and the only auger which will cut any kind of a knot without injury.

Hot-Blast Furnace Door.

In order to facilitate perfect combustion at a time when a fire is apt to smoke badly, that is, immediately after stoking, it is found advisable to admit hot air for a short time within the grate, in order to ignite the escaping unconsumed carbon. To meet this requirement, the James Reilly Repair & Supply Company, 230 West street, New York city, has produced a furnace-door liner having a large number of pins cast on the inner face, as shown in



the figure. This liner fits closely to the upper and side flanges of the door, but has an opening at the bottom whence the air, which enters the upper portion of the door at the top, may escape into the furnace. As this liner is intensely heated when in use, the air will absorb heat from the pins and thus enter the furnace in the heated state that is desired.

TECHNICAL PUBLICATIONS.

Business Features of Engineering Practice. By Alex. C. Humphreys, M.E., ScD., LL.D., President of the Stevens Institute of Technology. Pages 187. Hoboken: Published by Stevens Institute of Technology, 1905.

This very sparsely covered field of engineering practice has received a notable addition in the present work by President Humphreys. The importance of the subject should co-operate with the paucity of information to be found in print, in making the work exceedingly valuable. It consists in a series of excellent lectures before the institute delivered by President Humphreys and others, and covers a wide range of subjects coming under the general head indicated by the title.

Gas Engines and Producer-Gas Plants. By R. E. Mathot, M.E. Translated by W. B. Kaempffert, with a preface by Dugald Clerk. Pages 314 + 9; illustrations 152. New York: Norman W. Henley Publishing Company, 1905. Price $2.50.

This book covers a field which is as yet in part almost untrodden, and as a pioneer in the subject ought to have a great influence upon the development of the producer-gas outfits, which are now beginning to attract so much attention among engineers. It is an essentially unmathematical descriptive treatise of gas engines and gas producers, splendidly illustrated with diagrams of different types of machines and appliances, and intended as an operating guide for both designers and users in the construction and operation of both producer plants and gas engines. Valuable suggestions are given regarding the purchase of an engine, its installation, and questions and problems arising in connection with its care and operation.

Experimental Researches on the Flow of Steam Through Nozzles and Orifices. By A. Rateau. Translated by H. Boyd Brydon. New York: D. Van Nostrand Company, 1905. Price $1.50, net.

This little book consists of 76 pages of reading matter and is illustrated with four figures in the text and four folding plates illustrating the apparatus and curves of results from which the conclusions were drawn. The experiments were made at the time that Professor Rateau was engaged in the studies which finally resulted in the steam turbine bearing his name. They relate to the fundamental problem of steam turbine design, involving the proportions of nozzles and passages; and while distinctly a problem in physics, with all the care and precision which that implies, they were at the same time based upon an appreciation of practical considerations with regard to construction and operation. The importance of the book to the designers and builders of steam turbines can scarcely be overestimated.

Alternating Current Machinery. By William Esty, S.B., M.A., Lehigh University. Pages 412; figures 380. Published by the American School of Correspondence, Chicago, 1905. Bound in half-morocco. Price $3.75.

This book was written for the express use of practical men whose knowledge of mathematics extends no further than through the elementary principles of geometry and trigonometry; for this reason great pains have been taken to give great simplicity and clearness to all descriptions, explanations, and proofs. Graphical or geometric methods have been adopted wherever possible as giving a ready and easily understood explanation of various phases of electrical phenomena. The first 200 pages of the book are devoted to a description of various types of alternating machinery, together with measuring instruments for recording their operation. This part of the work is immediately followed by descriptions of synchronous motors, transformers, rotary converters and induction motors, together with chapters on switchboard and station appliances and apparatus. The illustrations are clear and well designed, and are of great value in following the text, while a good index at the rear of the volume places the work in the category of a handbook.

The Automobile Pocketbook. By E. W. Roberts, M.E. Pages 329; diagrams and illustrations 51. Bound in limp leather, 3 1-2 by 5 1-4 inches. Cincinnati, Ohio: The Gas Engine Publishing Company, 1905. Price $1.50, postpaid.

Designed to give clear and concise information on the general subjects of the operation and care of an automobile, this little book tells what to do in the case of an emergency, and, at the same time, includes a great deal of information on the subject of design of the various parts. In the interest of conciseness all reference to the historical side of the subject has been entirely omitted, and the book as a whole is intended to cover just exactly the points required by the user or designer of the machine, without any extraneous matter. It fits very handily in the pocket, and ought to find a ready field among all interested in the subject.

Gas Engine Design. By Charles E. Lucke, Ph.D. Pages 254 + 66; figures 145. New York: D. Van Nostrand Company, 1905. Price $3.00, net.

The purpose of this book being to present in a compact form the general principles underlying the design of gas engines, together with data taken from reliable sources and applicable to the use of those engaged in the design and construction of this popular form of prime mover, the inventive side of the design, which is treated at great length in nearly every previous work on the subject, has been entirely omitted, it being assumed that the reader is quite familiar with it. This leaves for this book simply the quantitative side of the design, with a discussion of the forces acting among the various parts, and the designs of the parts for the double purpose of transforming those forces into useful work, and of resisting their tendency to disrupt the elements of the engine. The work is divided into three parts, of which the first treats of power, efficiency and economy, the second of the forces in the engine due to the pressures of the gases and to inertia, and the third, which comprises nearly one-half of the book, to the dimensions of the engine parts. The higher mathematics have not been shunned in the treatment of the subject, though whenever it has been possible to obtain a simpler form of treatment this has been done. Illustrations of the various curves of inertia and effort diagrams are plentifully supplied, and add greatly to the character of the work. Figures and typography are excellent.

Text Book of Naval Engines and Machinery. By John K. Barton, Commander, United States Navy. Pages 581; figures, 248. Annapolis, The United States Naval Institute, 1904.

This work consists of two volumes, the first being the text and the second a volume of plates, 35 in number. It is intended as a text book for the use of students in the United States Naval Academy at Annapolis, where there is required only a sound general knowledge of such parts of the engineering field as are of special and practical use to the naval engineer, supplemented by a considerable amount of practical work in the engineering shops and in the management and care of the engines and boilers of the various torpedo craft, monitors, and steam launches available at that station. No mention has been made of the design of steam machinery, the particular function of the book being the construction, operation, management, and care of such machinery as is found on naval vessels. The work starts from a basis of definitions of work, energy, power, etc., traces the action of steam under varying conditions, and after a more or less extensive description of the various component elements of steam engines and other naval machinery, goes into the question of the indicator and its uses, curves of crank effort, and the subjects of propulsion and screw propellers. Considerable attention is paid to the question of engine balancing, as well as to that of the steam turbine. An appendix is added, giving naval regulations for the care of engines and boilers, together with tables of logarithms, mean pressures and the properties of saturated steam.

The book of plates includes drawings of the main engines of the armored cruisers *Washington* and *Tennessee,* together with outlines of a large number of parts; and some drawings of refrigeration and turbine installations, as well as a section of the Curtis turbine.

Marine Engines and Boilers; Their Design and Construction. A handbook for the use of students, engineers, and naval constructors. Edited by Leslie S. Robertson. Pages 744; figures 535. New York: Norman W. Henley Publishing Company, 1905. Price $9.00 net.

This handbook is based upon the excellent work in German of Dr. G. Bauer, engineer-in-chief of the Vulcan Works at Stettin, and has been translated from the second edition by E. M. & S. B. Donkin. Part I is devoted to the main engines, and is subdivided into sections upon the determination of cylinder dimensions, the utilization of steam in the engine, a discussion upon turning moment and balancing, the general arrangement of main engines, and details of main engines. Part II is devoted to the innumerable pumps used for various purposes on shipboard, including air pumps, circulating, feed, and auxiliary pumps, operated both independently and by direct drive from the main engine. Part III is devoted to shafting, resistance of ships and propellers, in the latter item going into both design and construction. Part IV is devoted to pipes and connections. Part V treats of steam boilers, and is divided into sections upon the generation of steam, cylindrical, locomotive, and water-tube boilers, funnels and lagging, forced draft, and boiler fittings and mountings. Parts VI and VII give a large amount of information upon measuring instruments and various details for the engine room and boiler room. About 80 pages in Part VIII are devoted to an extensive collection of the various mathematical tables usually found in works of this sort, while the book concludes with an appendix upon the Admiralty report on naval boilers in the British navy, and a very complete index.

The illustrations are unusually complete and splendidly chosen, many of them consisting of folders, giving both half-tones and line engravings of various engines of considerable note; and the information given in general being taken from actual cases of splendid construction, and referred to the cases specifically, ought to be of immense value to any one using the book. In this connection, we might mention an extended table giving particulars of cylinder sizes and ratios and general constructional features upon a large number of ships of various types, running from torpedo boats and destroyers up to the largest class of warships and transatlantic liners.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the Editor in this column. All communications must bear the name and address of the writer.

Q. 290.—We would be pleased to have you answer in your notes and questions column what is considered the best practice in balancing a four-crank triple expansion engine, and at what angles the cranks are usually placed. S. M. C.

A.—This is a very complex subject, and one to which there is no definite answer unless the exact details of the particular case in hand are known. We find, however, an example given in the 1904 edition of Seaton's *Manual of Marine Engineering* for a four-cylinder triple expansion engine, having cylinders 23 1-2 inch, 38 inch, 43 inch and 43 inch, with a stroke of 36 inches. The arrangement of these cylinders is, low pressure, high, intermediate, and low, beginning at the forward end. The sequence of the cranks is high-pressure, after low-pressure, forward low-pressure, intermediate pressure, and again high-pressure. The engine is balanced on the Yarrow-Schlick-Tweedy system, and from the diagram we estimate the angles to be as follows: From high-pressure to after low-pressure, 77 degrees; to forward low pressure, 96 degrees; to intermediate pressure, 77 degrees; and back to high-pressure, 110 degrees.

Q. 291.—Can you tell me the process through which the large calibre and other guns are finished; that is, how they obtain the polished brown with which they are finished? M. W.

A.—The Bureau of Ordnance of the Navy Department has furnished us with the following information and formula regarding the process of putting a finish on heavy guns in the naval gun factory:

Solution: 6 drops of tincture of iodine,
2 ounces of corrosive sublimate,
4 drachms spirits of nitre,
1 " tincture of steel,
1 " nitric acid, C. P.,
4 1-2 pints of pure water.

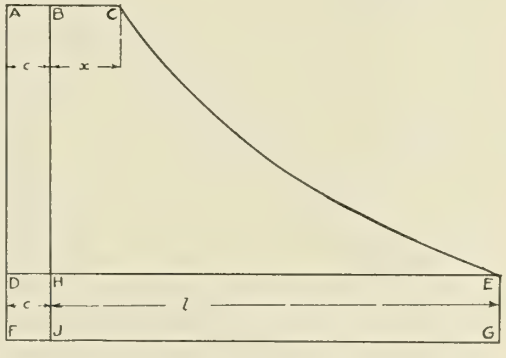
The guns must be made perfectly clean and free from grease by washing with a strong solution of potash, then washed with clean water, and wiped dry. Then apply the above solution with a soft, clean sponge. Let the solution remain on the gun for five or six hours until a dry coat of rust is formed. Then scratch brush the gun until the rust is scratched off; afterward wipe well with muslin cloth. Subsequently immerse the gun in a tank of boiling water and let it remain for about thirty minutes—the water to be kept boiling all the time that the gun is in it. After this the gun is allowed to cool until cool enough to bear your hand on it without discomfort.

This process is to be repeated until the gun has had at least five coats on it—sometimes it requires six coats—which is a matter of judgment. After the gun has had the required number of coats, rub well with the muslin cloth and apply a light coat of sperm oil with a flannel cloth.

It is best to use a circular wire brush attached to a flexible shaft, but it may be done with the ordinary file card held in the hand.

Q. 292.—Please answer this question and oblige. With steam at 60 pounds pressure, what would be the point of cut-off to expand down to atmospheric pressure? W. M. F.

A.—As this query does not give any information regarding the clearance of the engine in question we have assumed this to be 10 percent. Referring to the diagram, the clearance is represented by the distance AB, which we have designated c. The point of cut-off is C, and the percentage of stroke covered up to that point we have represented by x, the entire length of the stroke being l. Assuming that this steam is expanded adiabatically from a pressure at C of 60 pounds per square inch gauge to a pressure at E



equal to atmospheric pressure, or 14.7 pounds absolute per square inch, we find that the percentage of stroke covered up to C is about 15 1-2, which is obtained as follows:

Calling volume at point of cut-off $= v_c = c + x = 0.1 + x$, volume at end of stroke $= v_E = c + l = 1.1$, and pressures respectively p_c and p_E, and knowing that $pv^{10/9}$ is a constant for adiabatic expansion, we equate the value of pv at E, where both quantities are known, to that at C, where only the value of p is known. This gives us at once the value of v at C, thus:

$$p_C v_C^{10/9} = p_E v_E^{10/9}, \text{ whence } v_C = v_E \times \left(\frac{p_E}{p_C}\right)^{9/10}$$

$$= 1.1 \times \left(\frac{14.7}{60 + 14.7}\right)^{9/10}$$

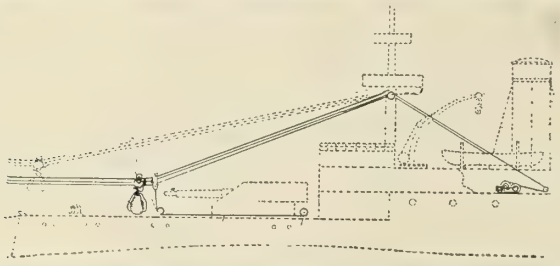
$$= 0.25465.$$

This gives $x = 0.15465$, or 15.465 percent.

SELECTED MARINE PATENTS.

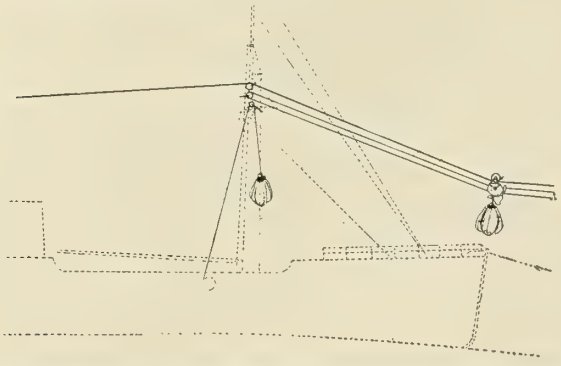
786,510. CONVEYING APPARATUS. THOMAS S. MILLER, SOUTH ORANGE, N. J.

Claim.—1. In combination, a supply-ship, a consuming-ship, a transit-rope connecting the two, a load-carriage, a forward transit-actuator on said consuming-ship and a backward transit-actuator exterior to said supply-ship.



2. In combination, a supply-ship, a consuming-ship, a transit-rope connecting the two, a load-support, a forward transit-actuator and a backward transit-actuator, both said actuators being on said consuming-ship.

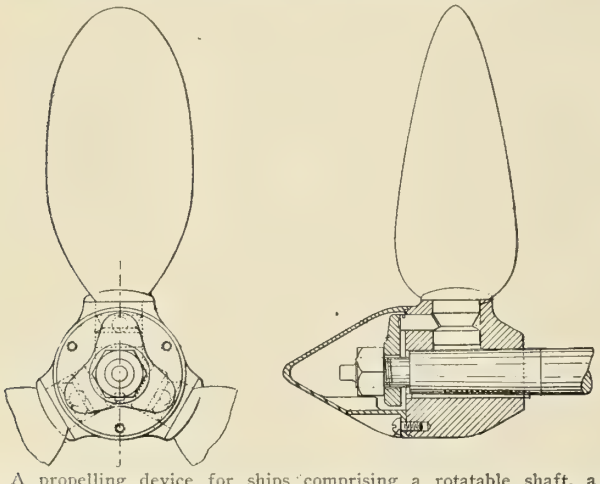
3. In combination, a supply-ship, a consuming-ship, a transit-rope connecting the two, a load-carrier, a supporting-rope, forward and backward transit-actuators and a supporting-rope tension-actuator; all of said actuators being exterior to said supply-ship.



5. In combination, a supply-ship, a consuming-ship, a ropeway connecting them, a load-carriage, a rope-drum mechanism on the consuming-ship and a sea-anchor participating in the control of the load during transit. Thirty-one claims.

787,084. APPARATUS FOR THE PROPULSION OF SHIPS. WM. T. DONNELLY, BROOKLYN, N. Y.

Claim.—4. A propelling device for ships comprising a rotatable shaft, a hub provided with sockets for blades mounted thereon, detachable blades mounted in said sockets, locking-pins adapted to secure said blades, a washer mounted upon the propeller-shaft, a tail-nut acting to force all parts axially along the propeller-shaft, and a cap or shield covering the tail-nut, secured to the hub and provided internally with projections which prevent the turning of the tail-nut while the cap is in place.



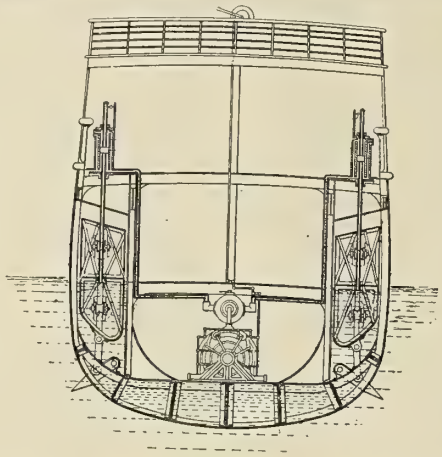
7. A propelling device for ships comprising a rotatable shaft, a hub provided with sockets for blades mounted thereon, detachable axially-adjustable blades mounted in said sockets, locking-pins, a tail-nut forcing the propeller axially upon the shaft, and means for securing the blades in their sockets by the axial thrust of the tail-nut along the shaft which presses the locking-pins inward. Seven claims.

787,811. FLOATING CURRENT-WHEEL. PHINEAS M. WARREN, NYSSA, ORE., ASSIGNOR OF ONE-HALF TO THOMAS J. HUFF, NYSSA, ORE.

Claim.—1. A floating current-wheel consisting of an axial shaft, a series of buoyant hollow buckets arranged parallel to the said shaft in circular series around the shaft and removed from the same, a circular brace connecting the buckets, and a series of radial hollow spokes communicating at their outer ends with the inner edges of the buckets and at their inner ends being connected with the central shaft and having openings as described whereby the hollow spokes form an integral part of the wheel and also act as drainage-tubes. Two claims.

787,182. WAVE-MOTOR. JOHN HUTCHINGS, LONDON, ENGLAND.

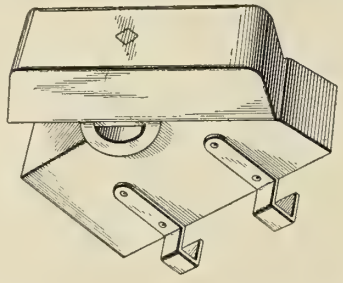
Claim.—6. In apparatus for forcing fluids a floating vessel—such as a ship—chambers therein accessible to the sustaining body of water and to the atmosphere, floats buoyed upon said water, chambers in said floats respectively imprisoning air and water therein, means of communication between



said water-chamber and the atmosphere and means for conveying the relative reciprocatory motions of the floating vessel and its contained floats to pistons and cylinders respectively connected thereto, pipes for conveying and valves for controlling the flow of the fluid being forced through said pipes to any suitable means for utilizing the same as motive power. Six claims.

787,294. BILGE-BLOCK. JOHN HICKLER, PASADENA, CAL.

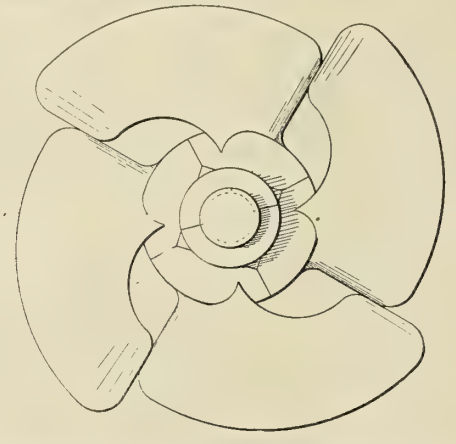
Claim.—The herein-described bilge-block consisting of a tilting member having a plane upper and lower face, a hollow semi-circular bearing-plate having end flanges, a plane upper surface, a semi-circular bearing-surface, said bearing-plate being secured to the tilting member by means of a bolt



passing through said block and through the plane upper wall of the bearing-plate, and a bearing member having a semi-circular recess therein to form a seat for the bearing-plate, the flanges at the ends of the bearing-plate serving to prevent sidewise movement of the tilting member. One claim.

787,745. PROPELLER. CALVIN T. FRIED, WEST ORANGE, N. J., ASSIGNOR TO FRIED ENGINEERING COMPANY, A CORPORATION OF NEW JERSEY.

Claim.—3. A propeller comprising a number of propeller-blades, each blade consisting of a face, representing the thrust area of the blade, and an outer marginal and curved edge on each blade, said edge being part of an increased spiral starting at the center of the propeller and said spiral representing the line of centrifugal force created due to the mass of the element upon which the blades act during their advance upon the same whereby said blades utilize the power expended in the creation of the aforesaid centrifugal force, by increasing the thrust parallel to the central axis of the propeller in like ratio.



14. The combination, with a propeller-shaft having a thrust-receiving shoulder, and a series of dovetail grooves, each dovetail groove extending in the direction of the central axis of the shaft and each dovetail groove having its central axis located at one side of said central axis of the shaft, to provide an arrangement of dovetail grooves eccentric with the said shaft, said shaft being provided with an annular groove and an annular shoulder, of a series of separately-constructed propeller-blades, a hub member at the root of each blade, a dovetail connected with each member registering with a dovetail groove, and a retaining-collar in said annular groove, comprising a pair of half-sections, and bolts for securing said half-sections together. Fourteen claims.